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THE DESIGN OF A CONSTANT STRESS CREEP TESTING UNIT

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THE DESIGN OF A CONSTANT STRESS CREEP TESTING UNIT

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Frank T. Shaver

THE DESIGN OF A CONSTANT STRESS CREEP TESTING UNIT

by

Frank T. Shaver

Lieutenant, United States Navy

Submitted in partial fulfillment of
the requirements for the degree of

MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING

United States Naval Postgraduate School
Monterey, California

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Frank T. Shaver

This work is accepted as fulfilling
the thesis requirements for the degree of
MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING
from the
United States Naval Postgraduate School

ABSTRACT

A lever and cam to produce constant stress was designed to be adapted to an existing tensile creep testing unit located at the U. S. Naval Postgraduate School. The lever and cam, when installed, produced a stress which varied plus or minus one percent during an extension of one inch in a 1.5-inch gage section.

The constant stress testing unit was then applied to creep testing a low carbon steel at 510°C, the results being compared with the results of tests made on the same material at the corresponding temperature under constant load.

The work was done at the U. S. Naval Postgraduate School from January to May of 1959.

The author wishes to express his appreciation for the assistance and encouragement given by his faculty adviser, Dr. Alfred Goldberg, Assistant Professor of Metallurgy.

The author also wishes to thank Mr. Robert Meyers for his assistance and encouragement, and Laboratory Technicians William D. Penpraze and Daulta Clark for their assistance. The author is also indebted to Chief Machinist V. R. Childers and the personnel of the Naval Postgraduate School Machine Shop for their cooperation and assistance during construction of the equipment.

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TABLE OF SYMBOLS AND ABBREVIATIONS

ε	Strain
$\dot{\varepsilon}$	Strain Rate
T	Absolute Temperature
σ	Stress Applied
σ_o	Initial Stress, Stress Immediately After Application of Loading Force
A	Cross-Sectional Area of Specimen
A_o	Initial Cross-Sectional Area
Δ Gage	Change in Reading of Dial Extensometer From Reading at Zero or Start Position
l	Length of Uniform Area Gage Section of Specimen
l_o	Initial Length of Uniform Area Section at Zero Strain
t	Time of Test
F	Specimen Loading Force
F_o	Specimen Loading Force Immediately After Initial Application of Load
α	Angle Position of the Weight Suspension End of the Lever From the Horizontal; Plus When Lever Above the Horizontal, Negative When Lever Below Horizontal
β	Angle Measured From Line Through Knife Edges to Tangent to Cam Curve at Point of Contact With Bearing
W	Force of Weight Load

1. Introduction.

Every day brings greater demand for new and better materials to be used at high temperatures. In addition there is a growing demand for more complete information about the behavior and expected performance of the metals and alloys already in use. In response, the fundamental study of creep and resistance to creep has been pursued by growing numbers of investigators and many theories have been offered to explain the mechanism by which creep occurs, to predict the behavior of materials, and to explain what properties contribute to creep resistance. Most of the theories view creep as a thermal activation process. Strain, ε, and/or strain rate, $\dot{\varepsilon}$, are presented as some function of stress, σ, temperature, T, time, t, and various constants. (1)[1] In general, then:

$$\varepsilon \text{ or } \dot{\varepsilon} = f(\sigma, T, t)$$

The application of any of these theories requires the use of the quantity, σ, called stress. The concept of stress involves the assumption of a homogeneous, continuous, and isotropic material. Most of the materials do not satisfy this assumption but experience has shown that at normal temperatures the solutions based on this assumption may be applied with very great accuracy provided the geometrical dimensions defining the form of a body of material are large in comparison with the dimensions of the crystals

[1]Numbers in parenthesis refer to the bibliography.

which actually make up the material. There is no evidence that the use of high temperatures decreases the accuracy of solutions based on the assumption of homogeneity, continuity, and isotropy.

In tensile creep testing great care is taken to ensure that no twisting or bending moment is imparted to the specimen, that is to say, the load is made to be uniaxial. The stress, then, may be found by the equation:

$$\sigma = \frac{F}{A}$$

where; F = specimen loading force
A = cross sectional area of specimen.

To facilitate the analysis of any set of data it is necessary to eliminate the effect of each of the independent variables in turn. In the case of creep test data it would be necessary to eliminate , in turn, the effect of stress and temperature. Unfortunately, in the conventional constant load technique for uniaxial tensile creep testing the area decreases continually as deformation progresses . Since the specimen loading force remains constant the stress continually increases as deformation progresses . Assuming the constancy of volume and assuming a uniform change in cross-sectional area in the gage section the stress will increase in direct proportion to the elongation as shown below.

$$\text{Volume} = \text{Constant} = A_o l_o = A l$$

$$A = \frac{A_o l_o}{l}$$

$$\sigma = \frac{F}{A} = \frac{F}{\frac{A_o l_o}{l}} = \frac{F l}{A_o l_o}$$

For example, if a specimen undergoes an extension of 25 per cent the stress will be increased by this same per cent. Analysis of creep test data from constant load tests are thus limited to comparing behaviors at equivalent strains under identical initial load conditions. It is only under constant stress (which implies a decreasing loading force) that a more useful analysis can be made over the whole range of creep straining.

Several methods for approximating constant stress have been presented in the literature. About 1910 Andrade made use of a weight of specially designed shape partially immersed in a liquid.(2). Caffyn(2)described two types of units for constant stress both in tensile and compressive testing. These consisted of levers bent at a ninety degree angle and pivoted either at the bend or at one end. Weights were used to supply the loading force. The range and accuracy were not indicated but,it appeared that these systems were very accurate for small amounts of deformation. Ward and Marriott(3)described a unit designed primarily for use with plastics. The loading force was supplied by weights suspended from a flexible strap or line. This flexible strap then passed over a cam of designed curvature such that as the specimen elongated the rotation of the unit effectively decreased the lever arm ratio

thereby decreasing the specimen loading force. Careful design of the cam maintained very nearly constant stress for elongation up to 50 per cent. Fullman, Carreker, and Fisher (4) described units which were also right-angle levers but pivoted on knife-edges at some point along one leg of the bent lever. These units maintained the stress with no variation more than 0.2 per cent for elongations up to 10 per cent.

None of these designs were suitable for adaption on the units at the United States Naval Postgraduate School. The design of a contour cam or a right angle lever either required space in excess of that available for adaption to the existing creep test stand or produced results over an unacceptably short range of elongation. A new approach was undertaken which would provide for the displacement of the point of weight suspension along the lever. Only a slight modification of the existing lever would be required and the system would be workable in the space available. This was preceded by a detailed study of the extension characteristics of the creep specimen in use.

In summary, then, what was required was to :

1. Determine the extension characteristics of the specimen in use.

2. Design a system such that as deformation progressed the ratio F/A remained constant.

2. Studies of specimen characteristics

The tensile creep testing machines at the United States Naval Postgraduate School were of the constant lever arm ratio type. A beam having three knife edges was mounted on the stand on the center knife edge. The force supplied by weights suspended from the knife edge at the long lever end of the beam was transmitted to the specimen at the knife edge at the short lever end. With known weights and the known lever arm ratio the desired specimen loading force could be obtained. This force remained constant throughout the range of movement of the beam unless weights were added or removed.

Once the specimen had been placed in the high temperature bath and the test had been started there were only two indications available to the observer to show what was happening to the specimen and to guide the changing of the loading force to give constant stress. These two indications were:

1. A mechanical dial extensometer attached to the specimen through the pin holes. See Fig.1.

2. Movement of the lever arm itself.

The type of specimen shown in Fig.1 had been chosen as the most desireable for high temperature study for several reasons. By use of a special jig it was possible with reasonable labor and expense to get very uniform and very nearly identical specimens.

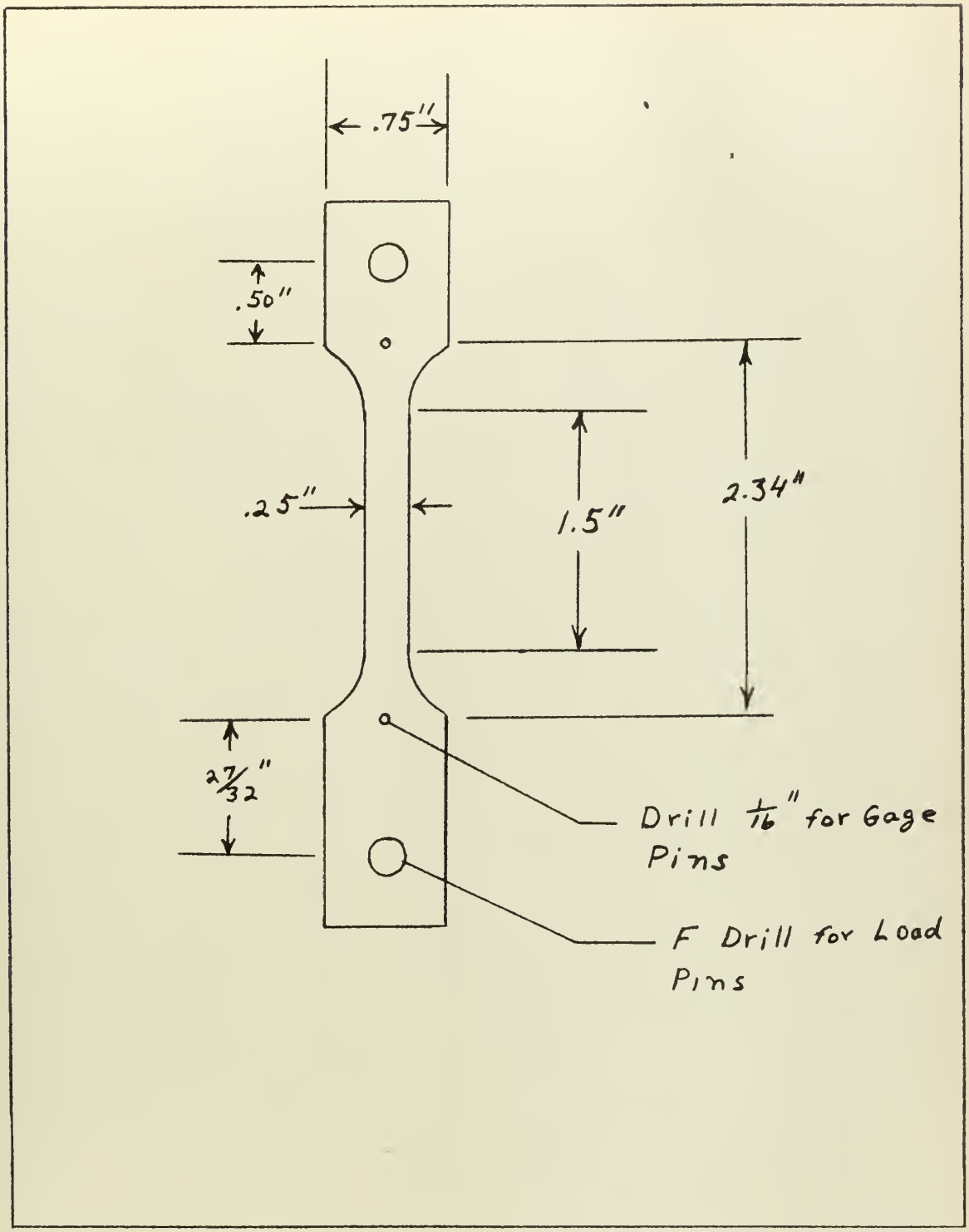


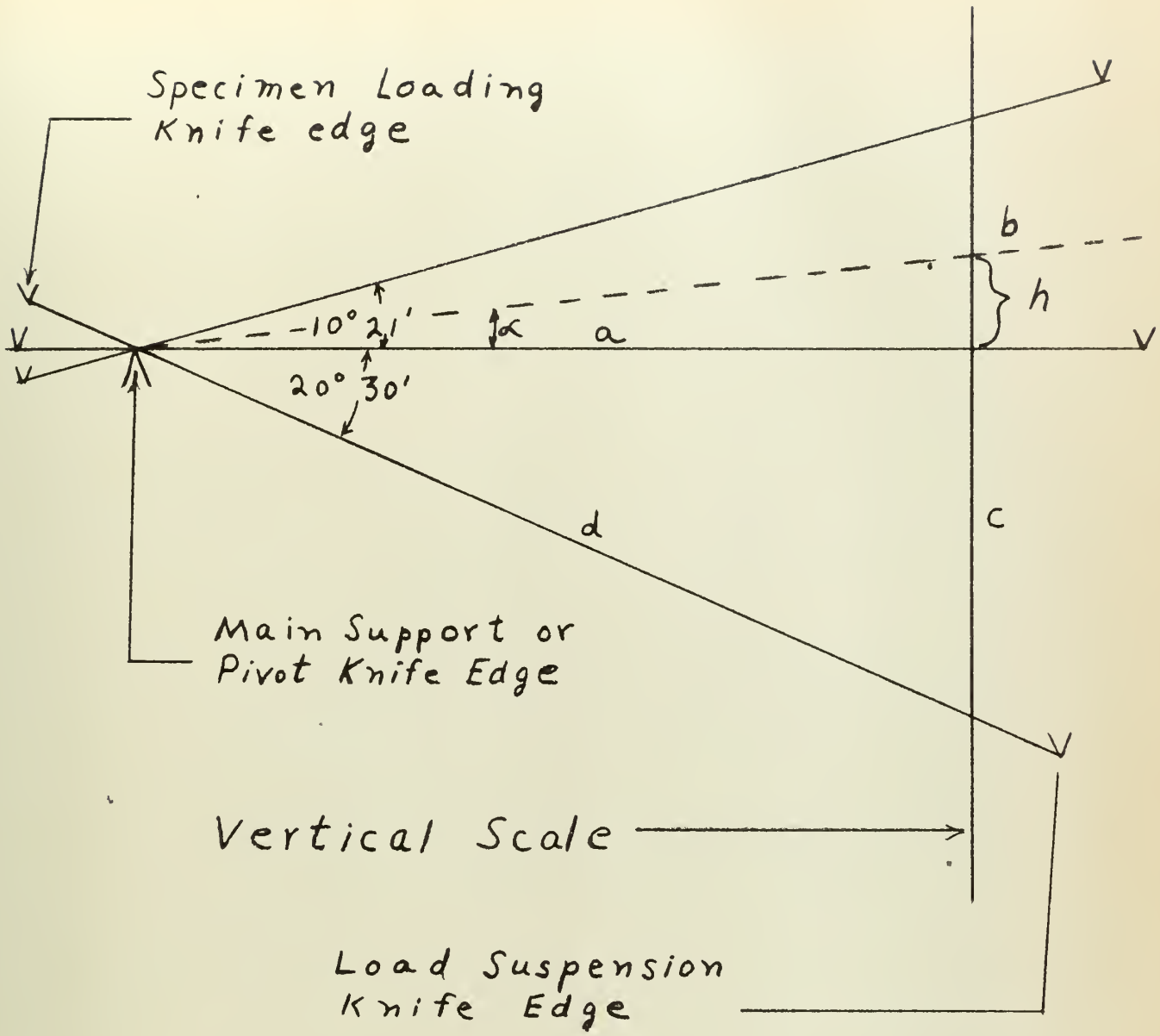
FIG 1. TENSILE CREEP SPECIMEN

The specimen had sufficient rigidity and designed clearances to make it easy to install in the liquid bath for testing.

Because weights providing a force in the true vertical were used as the means of loading it was necessary to study the movement of the lever arm as a change of angular position from the vertical or horizontal. The upper angle was limited by a lever arm stop while the lowest angular position was limited by the lever arm striking the edge of the top of the support stand. The angles were calculated from data taken in the following manner. The distance from the main support knife edge to a vertical scale was carefully measured with the lever arm in the horizontal position as indicated by a bubble level attached parallel to the line through the three knife edges. (See Fig. 2) Then the intersection of the vertical scale with the line containing the knife edges was recorded at the same instant the dial extensometer attached to the specimen was read during creep tests. The angular position, α, of the lever arm for a given change in dial extensometer, Δ gage, could be calculated by the equation:

$$\alpha = \tan^{-1} \frac{h}{30.19}$$

Where h is the distance along the vertical scale from the horizontal intersection of the line through the knife edges.



distances:

a = 30.19"
b = 5.44"
c = 11.50"
d = 32.30"

FIG. 2 LINE DIAGRAM OF SETUP FOR DETERMING α

Data for the calculation of the angle, α, was taken during creep straining of three specimens of low carbon steel at 605°C and four specimens of aluminum (2024-T3) at 485°C. Because of fracture of the specimens no data was obtained for values of Δ gage greater than 0.6 inches. The results of these tests are presented in Fig. 3. At the starting position α was found to be 10°21'. The lagging of the specimen gage reading increment, Δ gage, relative to the angle change in the early stages could have been due to:

1. Elastic deformation of the loading mechanism on initial application of load.

2. Some initial strain in the specimen between extensometer pins and load pins. (The loading force was transmitted to the specimen through stainless steel pins in the loading pin holes; see Fig. 1. The bottom loading pin is hooked in a jig in the bottom of the bath. The upper loading pin is hooked to a long stainless steel rod which, in series with a universal joint, a turnbuckle, and knife-edge block, takes the force from the specimen loading knife edge.)

Because the point of attachment of the specimen was such a large distance from the specimen loading knife edge the force transmitted could be taken as acting in the vertical regardless of the angle α. (See Fig. 4) This means that the elongation of the specimen could also be taken as movement in the vertical. For an ideal specimen, that is one in which all deformation occurs in the uniform area gage section and is the extension indicated by the value of Δ gage, the value of Δ gage would be a measure of the vertical displacement of the specimen loading knife edge. The

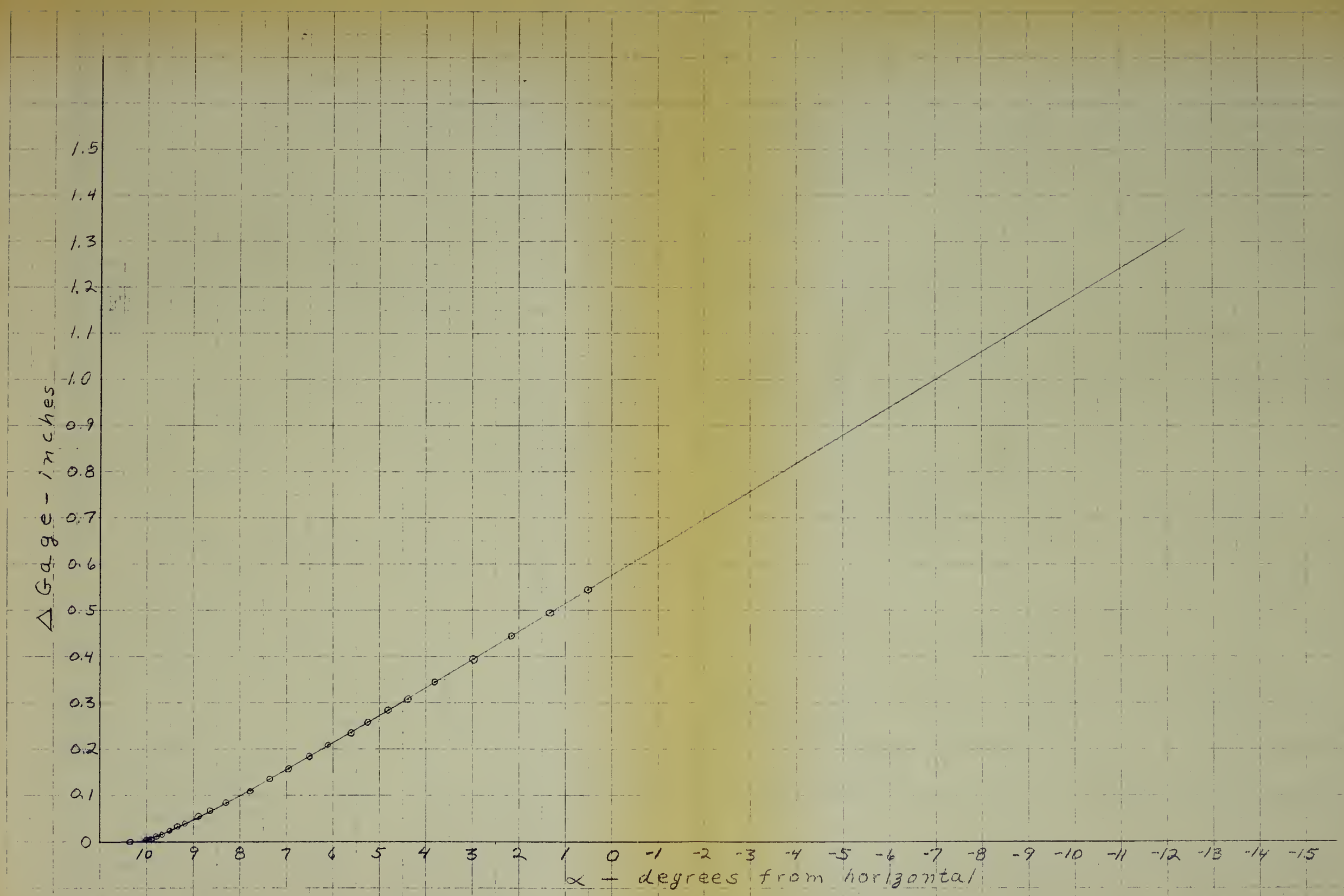
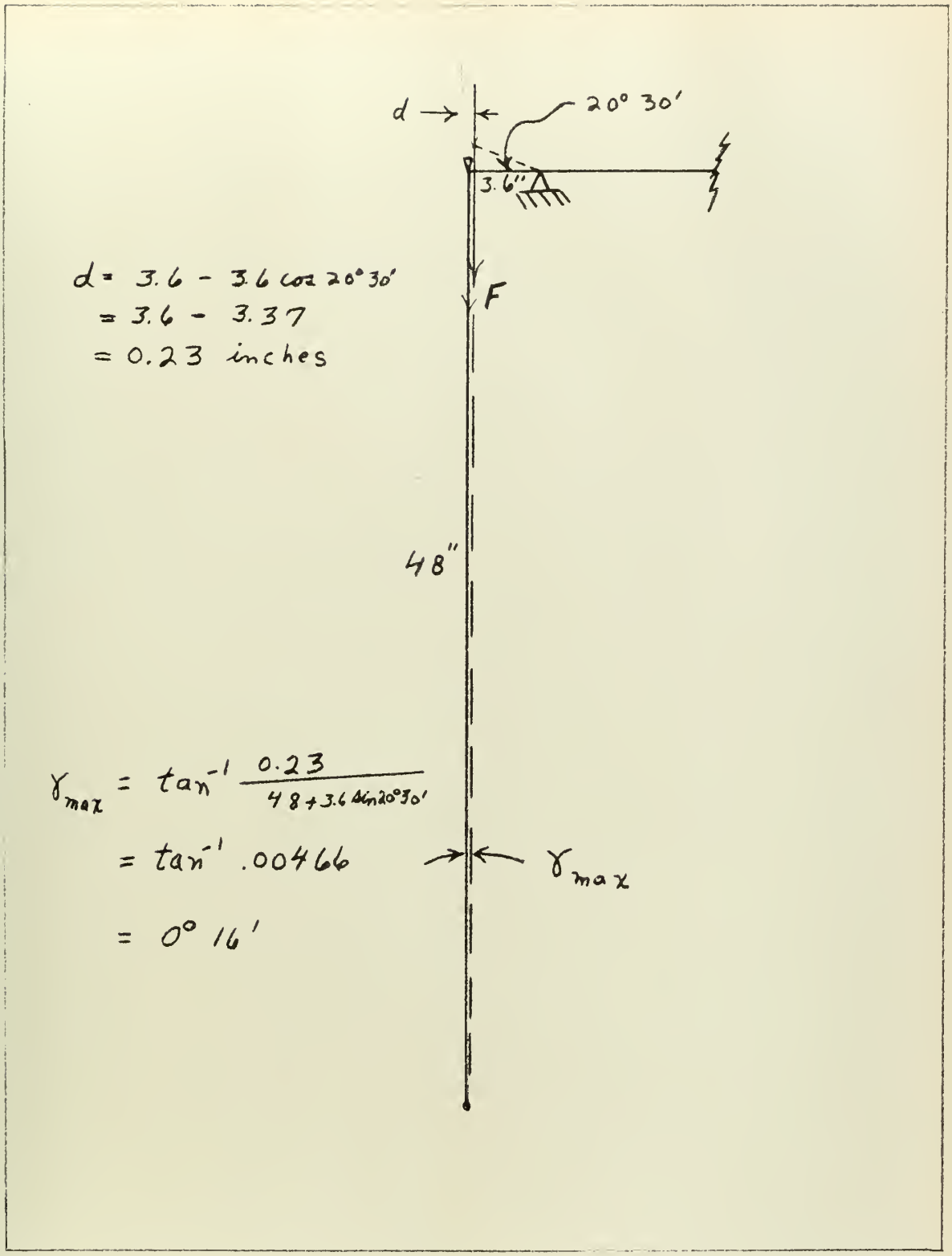


FIG. 3 PLOT OF Δ GAGE VERSUS ANGLE α



$d = 3.6 - 3.6 \cos 20°30'$

$= 3.6 - 3.37$

$= 0.23$ inches

$\gamma_{max} = \tan^{-1} \frac{0.23}{48 + 3.6 \sin 20°30'}$

$= \tan^{-1} .00466$

$= 0° 16'$

FIG. 4 VERTICAL CONSTANCY OF F

ideal relation between the values of Δ gage and the angle α would be as follows:

$$\Delta \text{ gage}_{\alpha=0^\circ} - \Delta \text{gage}_{\alpha} = 3.6 \sin \alpha$$

For small angles the relation would be linear and this is indicated in Fig. 3 as α approaches zero. In the absence of data beyond the horizontal position of the lever arm the linear relation in the vicinity of α equal zero degrees was extrapolated to the region of negative values of alpha. This undoubtedly was introducing a small error. From Fig. 3 the value of Δ gage for $\alpha = 0^\circ$ is 0.573. Then for $\alpha = -12^\circ$

$$\Delta \text{gage}_{\alpha=0^\circ} - \Delta \text{gage}_{\alpha=-12^\circ} = 3.6 \sin(-12^\circ)$$

$$\Delta \text{gage}_{\alpha=-12^\circ} = 1.321$$

Fig. 3 gives a value of Δ gage for $\alpha = -12^\circ$ of 1.303.

Consideration of the specimen design, with reference to Fig. 1, indicated that there was an uncertainty in the indication of extension available to the observer from the dial extensometer. The mechanical dial extensometer was actuated by the relative movement between two sets of stainless steel rods. By means of fixtures each set of rods was fastened to the specimen at one end of the gage section with pins serving as locators to ensure the same placement of the stainless rods on each specimen or on successive

placements on the same specimen in interrupted tests. For obvious reasons the extensometer pin holes could not be placed anywhere within the 1.5-inch gage section where every effort had been made to ensure uniform cross-sectional area. These pin holes had been placed at the extremities of the fillet radii at a distance of 2.34 inches. Could all the extension indicated by the dial safely be assumed to occur in the 1.5-inch gage section of uniform cross section? If not, was the actual extension predictable for each value of Δ gage?

Studies were undertaken to answer these questions. Fine reference marks were carefully scribed at the center and ends of the uniform area gage section of standard specimens. The distance between these fine scribe marks was carefully measured by means of a traveling microscope graduated to .001 centimeters or .00039 inches. In addition, the distance between the extensometer pin holes was measured with the traveling microscope and the dimensions of the uniform area section were measured with a micrometer to determine the cross-sectional area. The specimens were then strained in a constant load tensile creep test with frequent interruptions for measurements. The distance between scribe marks was measured so as to give the length of the uniform area section at each stage of the deformation. However, it was necessary to compensate for the fact that in spite of the greatest care it was not always possible to

place the scribe marks at the ends of the uniform area section exactly 1.5 inches apart. The measured distance between the marks at each stage of deformation was divided by the distance between the same marks before any deformation. This value was then multiplied by 1.5 to give the length of the uniform area section at each stage of deformation. The change in distance between extensometer pin holes gave the value of Δ gage for each stage of the deformation.

Four specimens of aluminum (2024-T3), two at 450°C, two at 485°C and two specimens of recrystallized low carbon steel at 605°C were used. The results of these tests are presented graphically in Fig. 5. The response of an ideal specimen, as defined before, is shown by the dashed straight line in Fig. 5. The results indicated that the specimens used deformed differently than an ideal specimen. However, these results indicated that the actual deformation and therefore the length, at any given value of Δ gage, was predictable, at least to within plus or minus one percent. Also, this appeared to be a characteristic of the specimen design and was independent of material and temperature within the limits tested.

By assuming that the volume of the uniform area section remained constant during creep deformation, the cross-sectional area of the specimen could be calculated for any value of Δ gage from:

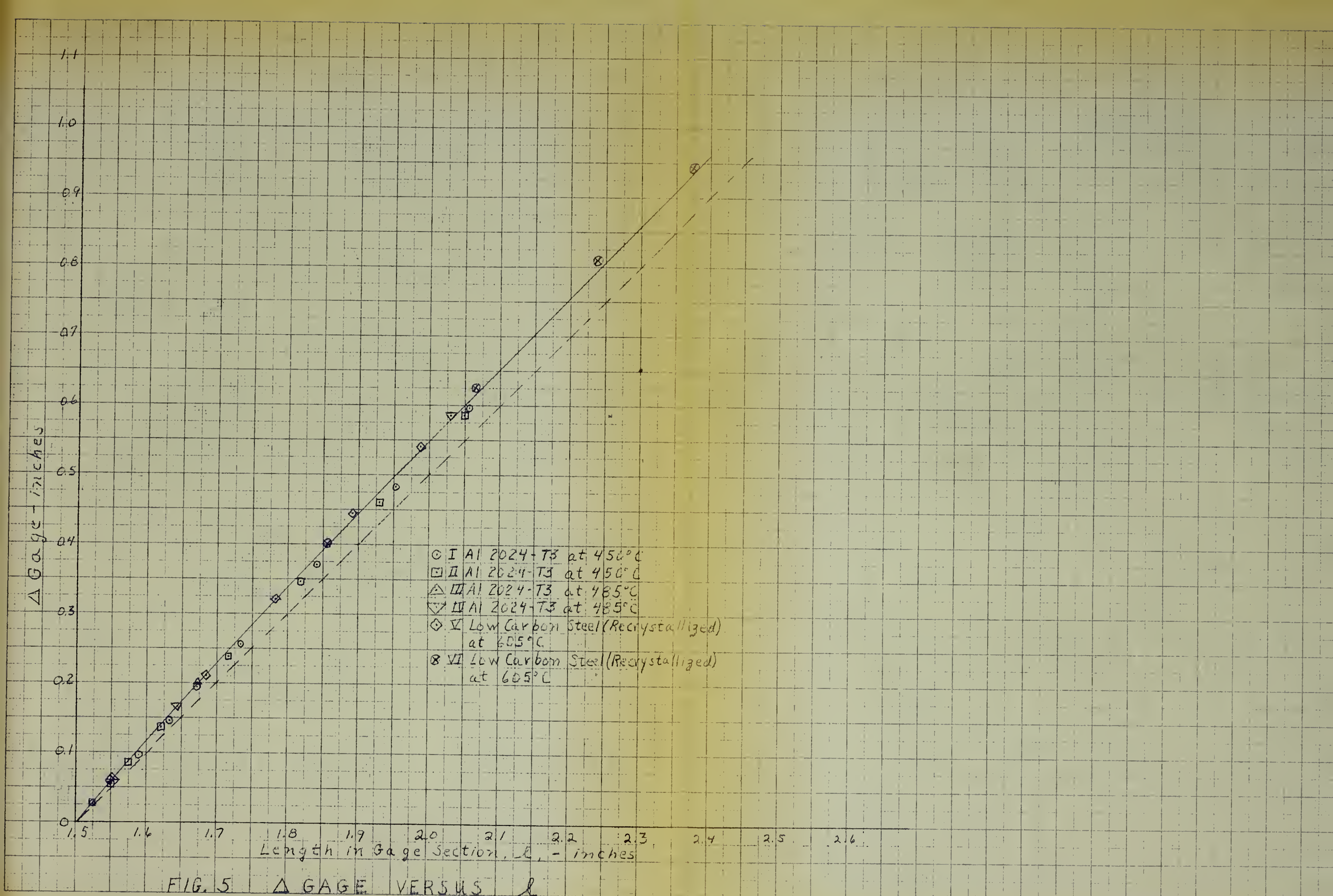


FIG. 5 Δ GAGE VERSUS ℓ

Since the specimen was manufactured so that ℓ was always 1.5 inches the ratio A/A_o could be found for any value of $\triangle$ gage by dividing 1.5 by the corresponding value of from Fig. 5. These calculations were made and the results plotted as the curve in Fig. 6. The maximum extension achieved by a specimen from which data for the curve came was .945 inches. It was necessary to extrapolate for values of $\triangle$ gage greater than .945 inches.

The cross-sectional area was calculated from the dimensions at each interruption. After the deformation had progressed to some extent, it was found that the cross section did not remain uniform throughout the gage section. The maximum and minimum cross-sectional areas in the gage section were determined. These values of cross-sectional area were reduced to a ratio A/A_o and were plotted in Fig. 6. The specimens which no longer had uniform area in the gage section produced a range of values of A/A_o for each stage of deformation. These ranges of A/A_o are indicated in Fig. 6 by two values of A/A_o connected by a dashed line.

The values of A/A_o for the actual measured areas followed the predicted curve quite well in the lower values of $\triangle$ gage but showed quite wide scatter and ranges above 0.3 or 0.4 inches of extension. It was certainly expected that variations would develop during creep deformation, particularly for those specimens exhibiting ductile failures. It had been hoped that such wide variations would not have developed so early in the deformation. There was not time to determine the effect of initial stress and temperature

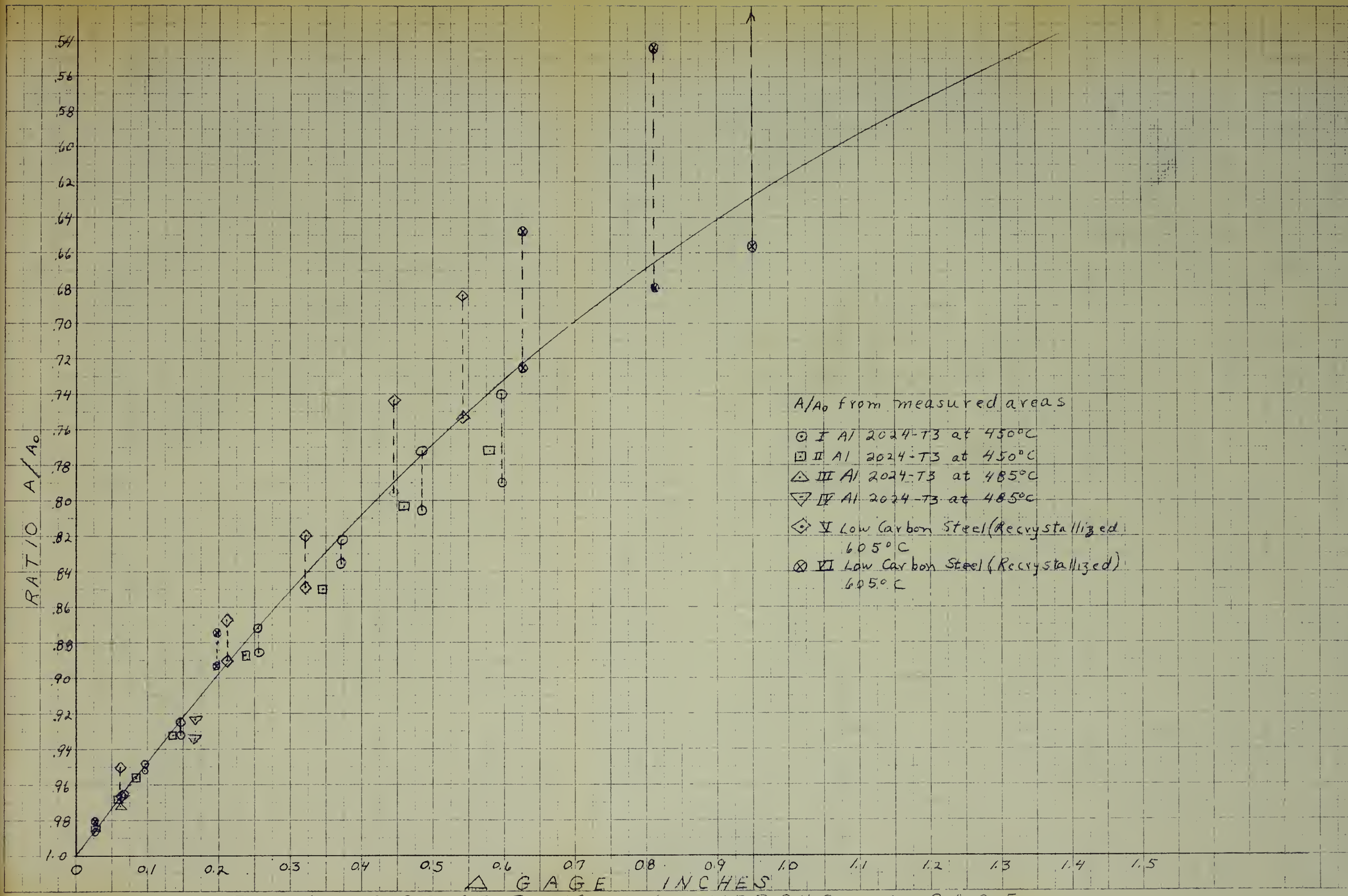


FIG. 6 PLOT OF RATIO A/Ao VERSUS Δ GAGE

on the magnitude of variation in area.

3. Design of the constant stress system.

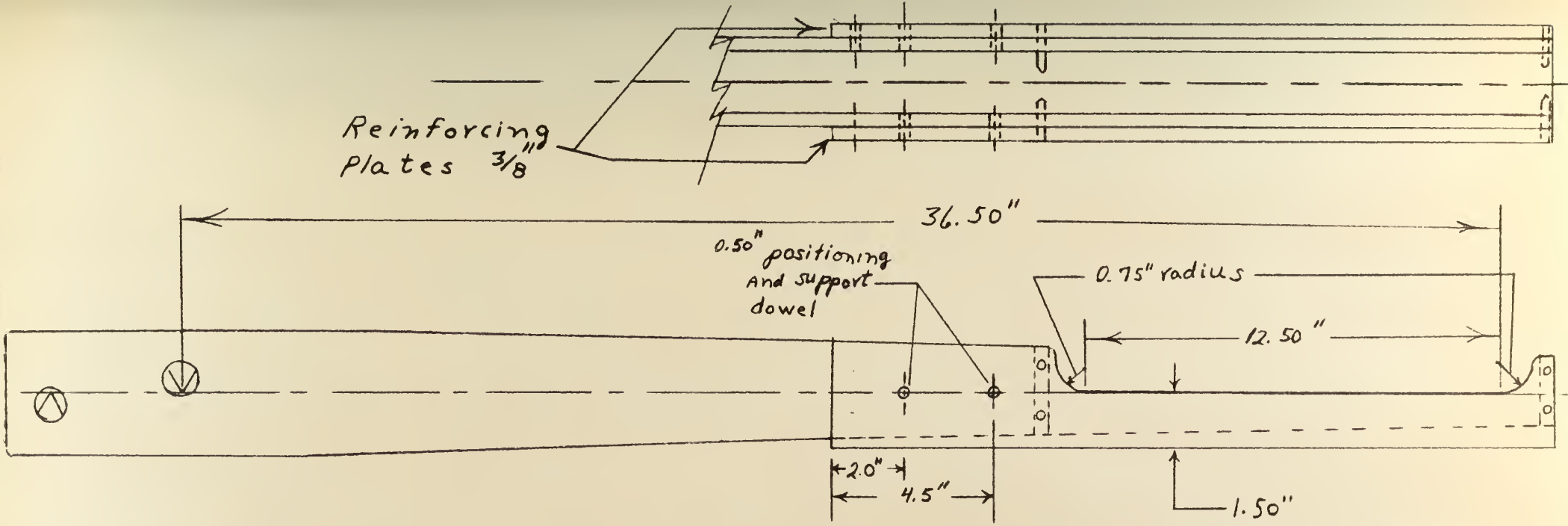
The local necking indicated by the variation of the minimum cross-sectional area from that predicted by assumption of constant volume, amounted to as much as one percent for Δ gage up to about 0.25 inches and increased rapidly thereafter to as much as 15 percent at Δ gage of 1.0 inches. It might have been questioned whether it was feasible or worthwhile to continue efforts to design a system for constant stress which did not take into account this area variation. Such a system would require either a continuous source of area measurement or an extreme amount of testing to determine the curve of A/A_o through the minimum area values. This minimum area curve would certainly vary with material and with temperature and stress as well. The curve of A/A_o as shown in Fig. 5 acceptably predicted the actual A/A_o as determined by measurement for a Δ gage up to 0.25 inches and gave a good average of the scatter above that point. Therefore, it was decided that it would be worthwhile to attempt to follow this curve with the specimen loading force, F. If this could be done then stress, constant to within one percent for the low values of delta gage, and a "best average" stress would be held constant for greater deformations.

The least complex and least expensive system that could be devised which seemed capable of approaching the desired accuracy, within the given space allowances, was a bearing-mounted weight suspension element with a cam for

positioning the point of contact along the lever arm. This allowed continued use of weights and lever arm advantage for supplying the specimen loading force but provided for changing both the lever arm ratio and the force transmitted to the lever arm by the weights as means for adjusting F to match A so that F/A would remain constant.

An existing lever arm was modified to provide a track for the weight supporting bearings. (See Fig. 7) This track for the weight support bearings was a machined and ground surface along the line through the two knife edges. This meant that except for the rolling friction the bearings would transmit a force to the lever arm perpendicular to the line of knife edges. The magnitude of the force would be a function of the angle of the lever arm from the horizontal, the weights used, the angle of the cam relative to the line of the knife edges, and the coefficients of friction involved. The use of precision bearings reduced the coefficient of friction to .001 to .0015.(5) This would cause an error of 0.39 percent or less (See Appendix I) in the stress produced; therefore, the friction was ignored in the following design calculations.

The weight suspension element consisted of a weight suspension bearing block which is shown in Fig. 8, and a bearing shaft, two weight pivot bearings, two cam bearings, and four lever arm contact bearings, shown in Fig. 9. One-inch diameter bearings were selected for the cam bearings and lever arm contact bearings. This suspension of



Reinforced after end of creep testing machine weight lever by securing 3/8" steel plates, as shown, to existing weight support beam.

Beam Cross Section in Region of Bearing Track

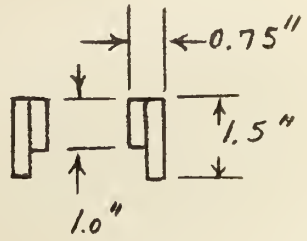
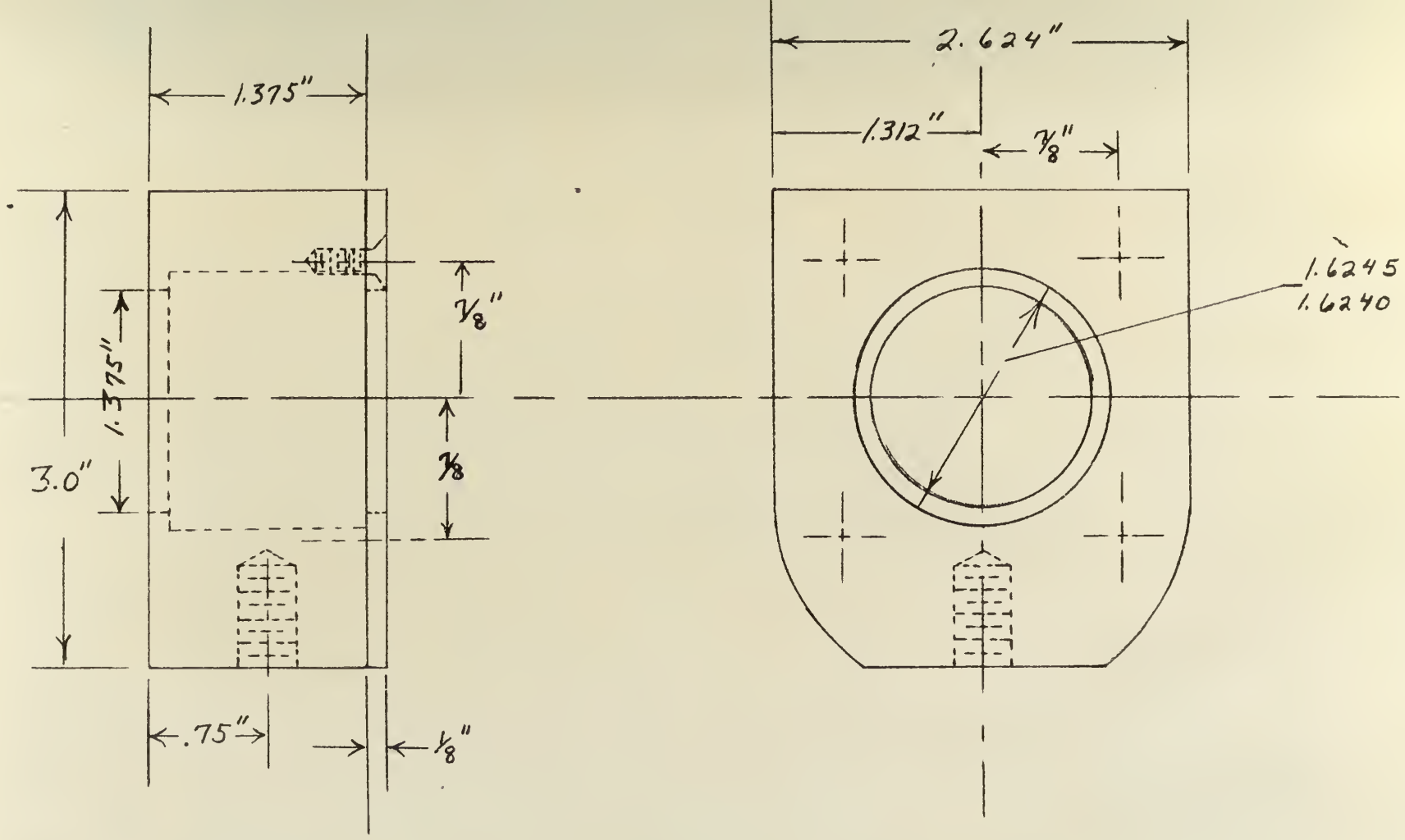


FIG. 7 MODIFIED LEVER



Scale 1" = 1"

1. Cover plate held by four countersunk screws.

2. Tapped hole in botton for weight rod.

FIG. 8 WEIGHT SUSPENSION BEARING BLOCK

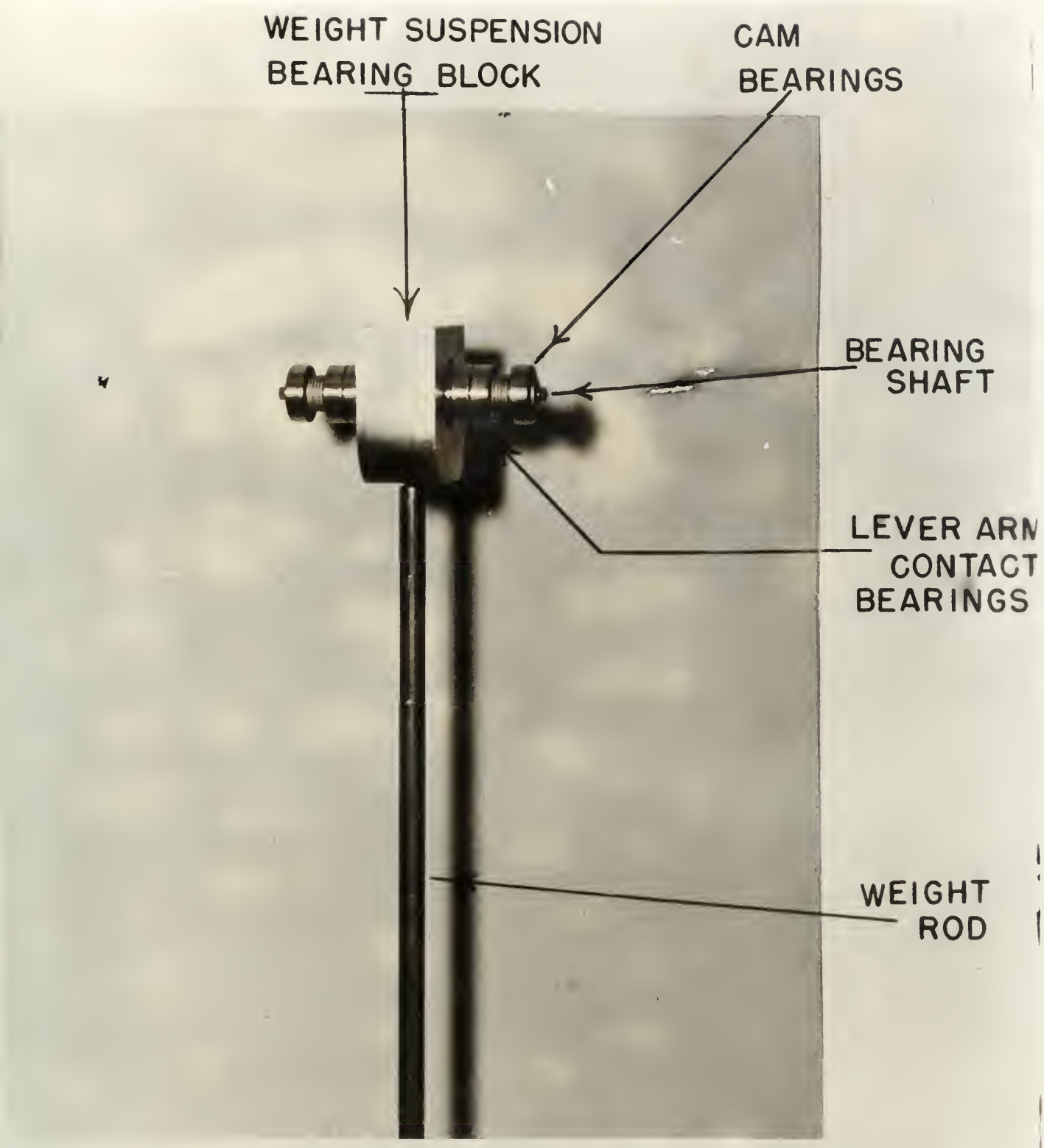


FIG. 9 PHOTOGRAPH OF WEIGHT SUSPENSION ELEMENT.

the weights ensured that the weights would hang directly under the suspension element exerting a vertical force on the bearing shaft. Fig. 10 and Fig. 11 show the complete assembly.

It was now possible to proceed with the design of the cam to position the weight along the lever arm. When the lever arm was above the horizontal, that is, near the starting position, the situation would be as shown in Fig. 12. The vertical force, W, of the weights could be resolved into components perpendicular to the bearing track and along the track, as illustrated in Fig. 13. The component along the track must be resisted by the reaction of the cam. Since the coefficient of friction is so small, the reaction of the cam on the cam bearing and bearing shaft is taken as perpendicular to the tangent to the cam curve and passes through the center of the bearings and shaft. This reaction may be resolved into components along and perpendicular to the bearing track of the lever. (See Fig. 14) The component of the cam reaction along the track must be equal to $W \sin\alpha$. Since

$$\frac{cb}{ca} = \tan\beta \qquad \text{(See Fig. 14)}$$

then

$$ca = \frac{cb}{\tan\beta}$$

That is, the component of the reaction of the cam perpendicular to the bearing track would be equal to:

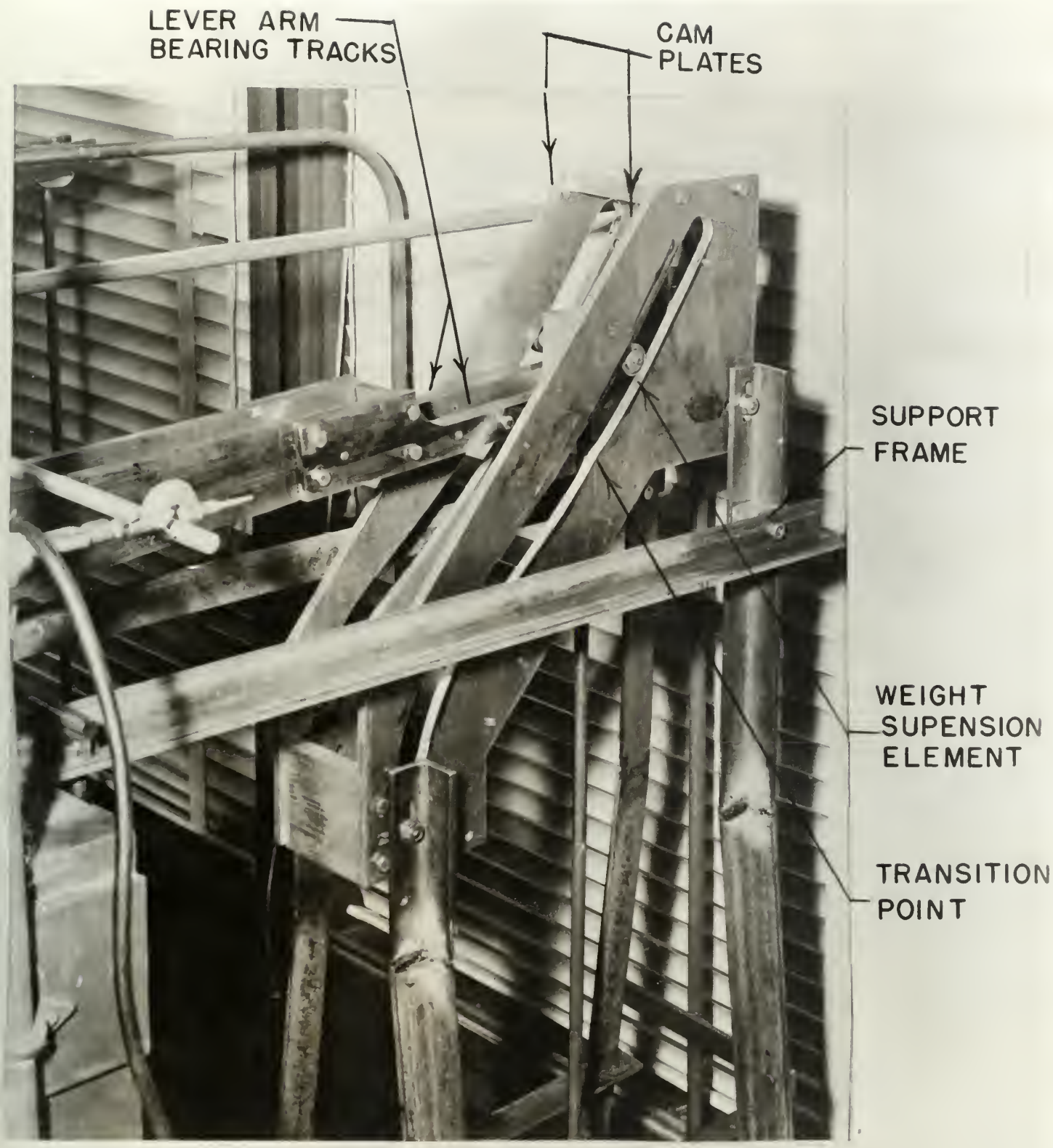


FIG. 10 PHOTOGRAPH OF LEVER, CAM, AND WEIGHT SUSPENSION ELEMENT.

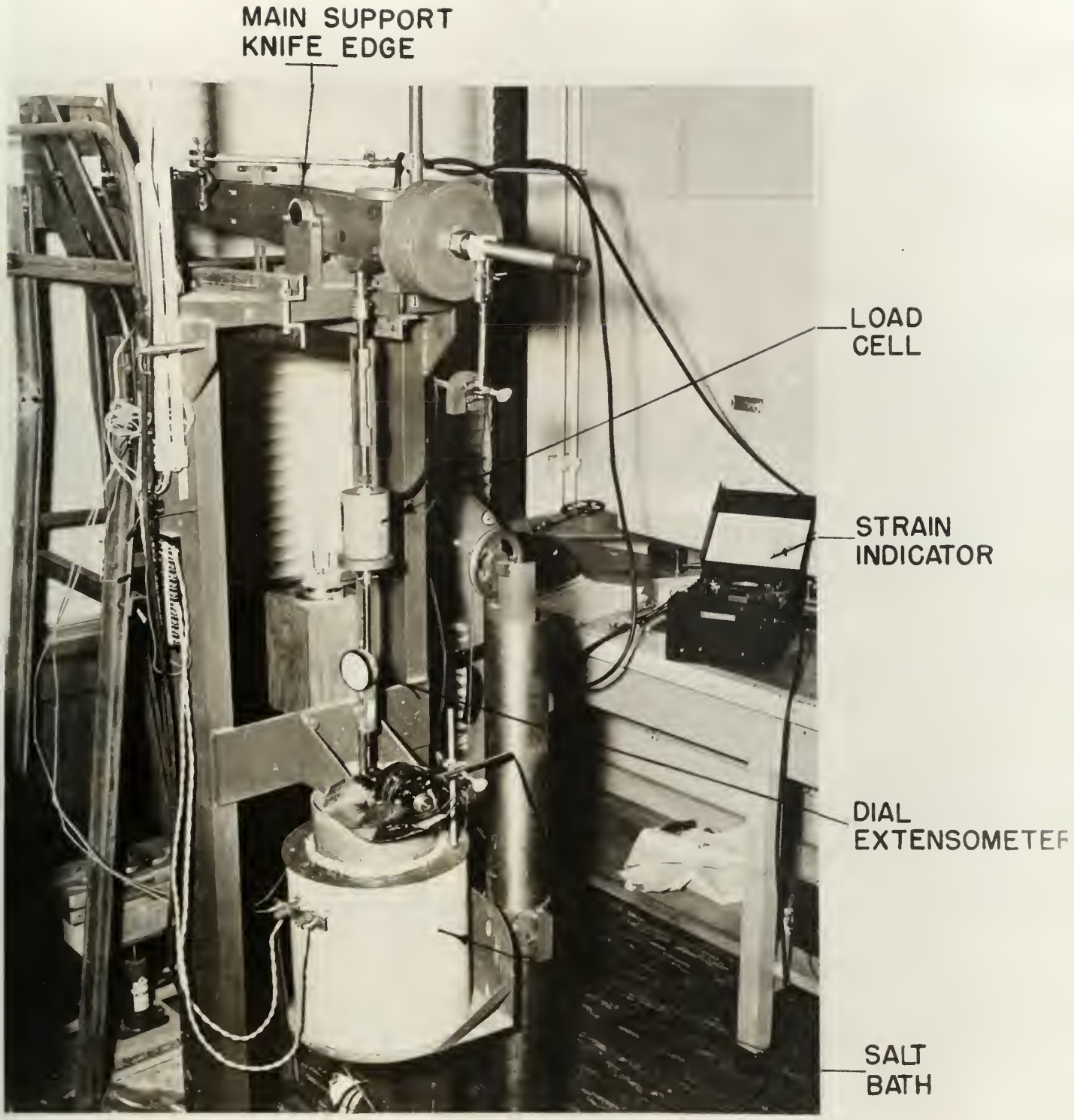


FIG. 11 PHOTOGRAPH OF COMPLETE TEST SETUP.

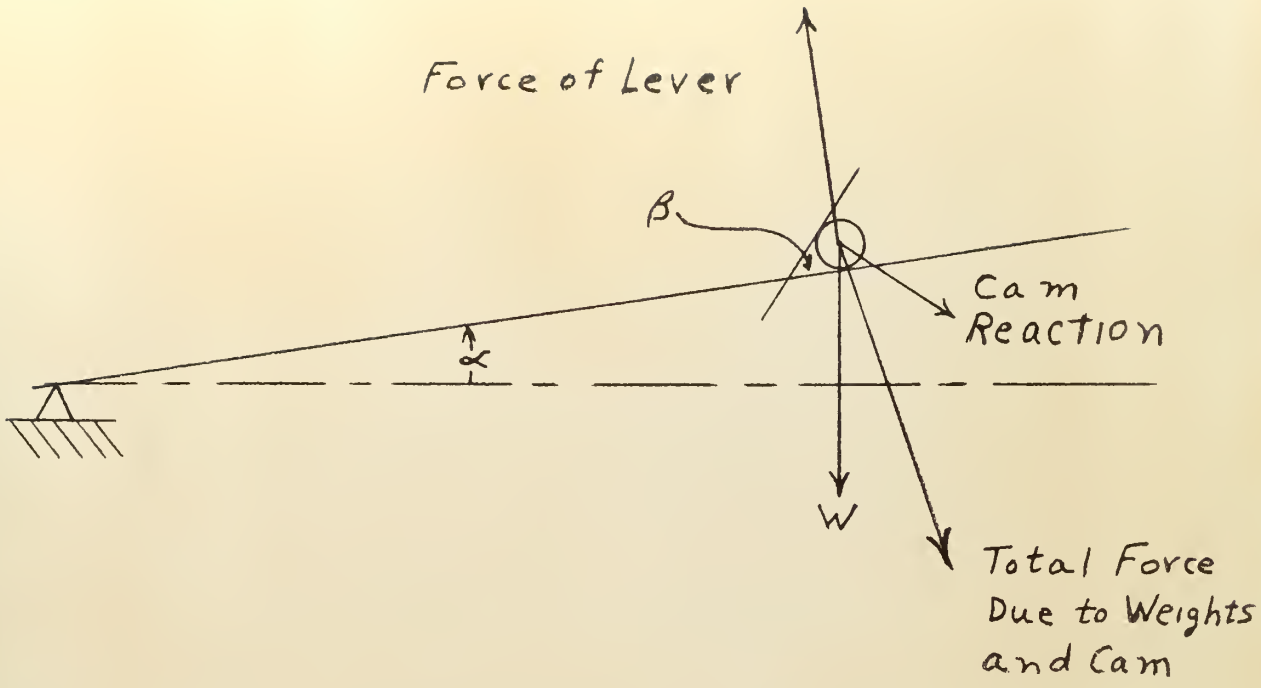


FIG. 12 FORCES ON BEARING SHAFT

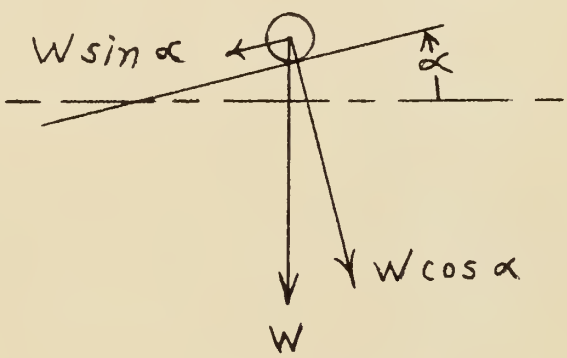


FIG. 13 RESOLUTION OF FORCE W

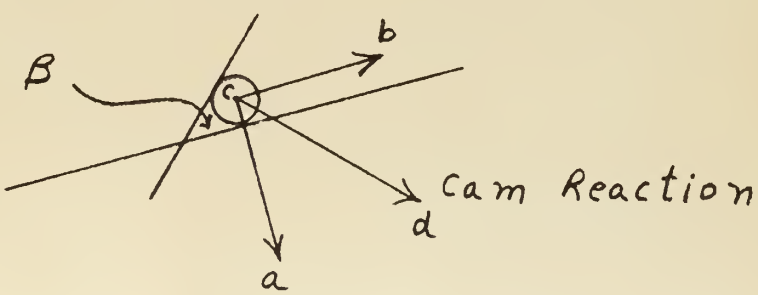


FIG. 14 RESOLUTION OF CAM REACTION

$$\frac{W \sin \alpha}{\tan \beta}$$

The total force transmitted to the lever arm through the bearings at the point of contact would be:

$$\text{force} = W \cos \alpha + \frac{W \sin \alpha}{\tan \beta}$$

$$= W\left(\cos \alpha + \frac{\sin \alpha}{\tan \beta}\right) \qquad (1)$$

Define x as the distance of the point of bearing contact on the lever from the main support, or pivot, knife edge. The moment, M, about the pivot knife edge of the total resultant force would then be:

$$M = x\, W\left(\cos \alpha + \frac{\sin \alpha}{\tan \beta}\right) \qquad (2)$$

The specimen loading force, F, acting through its lever arm would have to resist with a moment of equal magnitude. The specimen loading knife edge is at a fixed distance of 3.6 inches from the main support knife edge. However, as pointed out before, the force, F, could be taken as acting vertically for all positions of the lever arm. This meant that the resisting moment of the specimen loading force would vary with F and with α.

$$\text{moment} = F\ 3.6 \cos \alpha \qquad (3)$$

Equating the moments, equations (2) and (3)

$$F\ 3.6 \cos \alpha = W x \left(\cos \alpha + \frac{\sin \alpha}{\tan \beta}\right)$$

or

$$F = \frac{W x}{3.6}\left(1 + \frac{\tan \alpha}{\tan \beta}\right) \qquad (4)$$

This force, F, is the force which must be made to vary according to the change in cross-sectional area during creep deformation. The variables available for forcing F to the desired value are x and β. At α equal to zero degrees equation (4) reduces to:

$$F = \frac{W x}{3.6}$$

assuming that $\beta \neq 0^\circ$ when $\alpha = 0^\circ$.

It was anticipated that a distance of 29 inches for x at alpha equal to zero degrees would place the cam in an acceptable position along the lever. This was established as the initial design position of the point of contact for alpha equal to zero. Some such selection was necessary to establish a starting point for calculations. This selection established:

$$F_{\alpha=0} = \frac{W\ 29}{3.6} \qquad (5)$$

Then, for constant stress, at some other value of alpha

$$\frac{F}{F_{\alpha=0}} = \frac{A}{A_{\alpha=0}} \qquad (6)$$

By consolidation of data, Fig. 3 and Fig. 6, it was possible to obtain a plot of the ratio A/A_o versus the angle alpha, which is presented in Fig. 15. Since

$$\frac{A}{A_{\alpha=0}} = \frac{A/A_o}{A_{\alpha=0}/A_o}$$

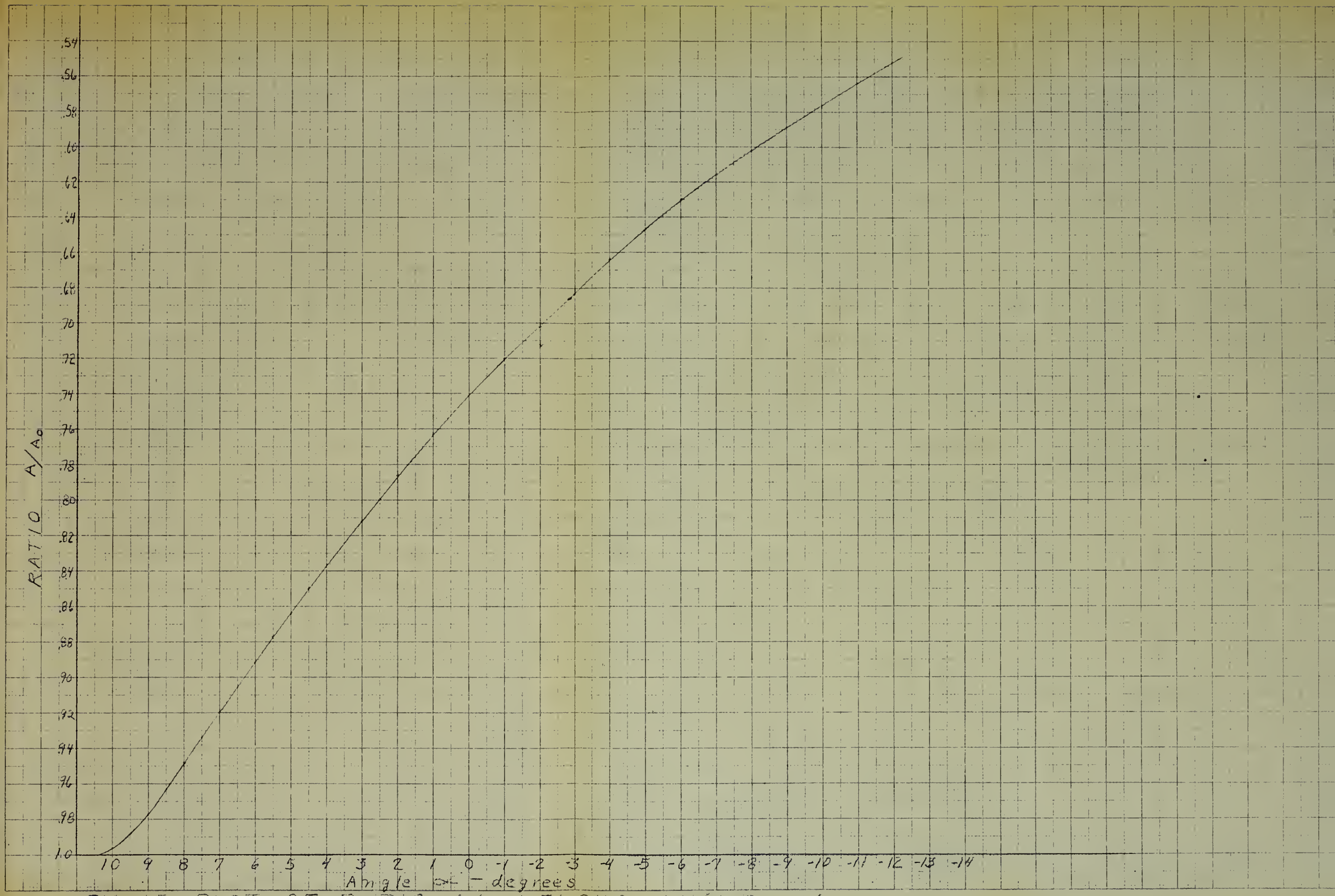


FIG. 15 PLOT OF RATIO A/A_o VERSUS ANGLE α

then

$$\frac{F}{F_{\alpha=0}} = \frac{(A/A_0)}{(A_{\alpha=0}/A_0)} \tag{7}$$

and, substituting from equations (4) and (5)

$$\frac{\frac{Wx}{3.6}\left(1+\frac{\tan\alpha}{\tan\beta}\right)}{\frac{W\,29}{3.6}} = \frac{(A/A_0)}{(A_{\alpha=0}/A_0)} \tag{8}$$

From Fig. 15, $A_{\alpha=0}/A_0 = .741$. Equation (8) then reduces to:

$$x\left(1+\frac{\tan\alpha}{\tan\beta}\right) = (A/A_0)\frac{29}{.741}$$

and

$$x = \frac{(A/A_0)\ 39.13}{1+\frac{\tan\alpha}{\tan\beta}} \tag{9}$$

For positions of the lever arm below the horizontal, equation (9) is valid if alpha is read as a negative acute angle. (See Appendix II) Assuming a value of alpha reduces the variables in equation (9) to two and x becomes a function of beta. By assuming a number of values of beta, corresponding values of x were obtained. For example:

at $\alpha = 1°$ $\quad A/A_0 = .76$

$$x = \frac{(.76)(39.13)}{1+\frac{\tan\alpha}{\tan\beta}} = \frac{29.75}{1+\frac{.0175}{\tan\beta}}$$

then: for $\beta = 45$ $\quad x = 29.23$ inches

for $\beta = 46$ $\quad x = 29.26$ inches

for $\beta = 44$ $\quad x = 2920$ inches

By repeating this for different values of alpha throughout the possible range of values for alpha, sets of x distances and corresponding tangent angles, β, were calculated. For any given value of the angle, α, there were an infinite number of values of x with corresponding values of beta which would result in the correct moment in the lever arm. However, there were conditions which limited the selection of the values and determined the desired solution. These were:

1. The cam curve must be smooth and continuous.
2. The cam curve must provide for the correct change in x as alpha changes.
3. The cam curve must assure that the value of x at alpha equal to zero is the designed value.

The change of x, dx, with a change in alpha, $d\alpha$, is a function of x, α, and β. To determine an approximately correct value of x for the next value of alpha to be considered the approximate relation

$$\Delta x \approx \frac{x \, \Delta \alpha}{\tan \beta}$$

which may be obtained from the geometry of the situation, was used. Knowledge of an approximate value of x for the next value of alpha to be considered avoided making an unnecessarily large number of calculations from equation (9) for each value of alpha to be considered.

The desired solution was obtained by a graphical method of tangent curve plotting. A number of small lineal elements, each representing a value of x and its corresponding angle beta, were laid out for many values of alpha

throughout the possible range of angles. The plotting method is shown schematically in Fig. 16. A reference point representing the pivot knife edge was established, the values of x were laid out full scale, and by allowing for the one-inch diameter bearings, the lineal elements representing a value of x with its corresponding value of angle beta were drawn in their true position relative to the pivot knife edge. The known starting point was at alpha equal to zero where the value of x had been selected. Although beta is indeterminate at alpha equal to zero, it was actually possible to use this as the starting point for the cam curve. This starting point is represented by the point P in Fig. 16. To arrive at the lineal elements for the next value of alpha at the correct distance x and the correct slope, or angle beta, the cam curve must follow very closely to the segment PB. To arrive at the correct lineal element for the next higher value of alpha the cam curve must follow very closely the segment BC, and by proceeding on to other values of alpha the entire cam curve was built up both above and below the horizontal position of the lever arm.

It is to be noted that a transition occurs at alpha equal to zero degrees. For positions of the lever arm above the horizontal the contact with the cam is made on the upper half of the bearing. As alpha goes through zero the component of W along the lever arm bearing track reverses direction. This means that the point of contact

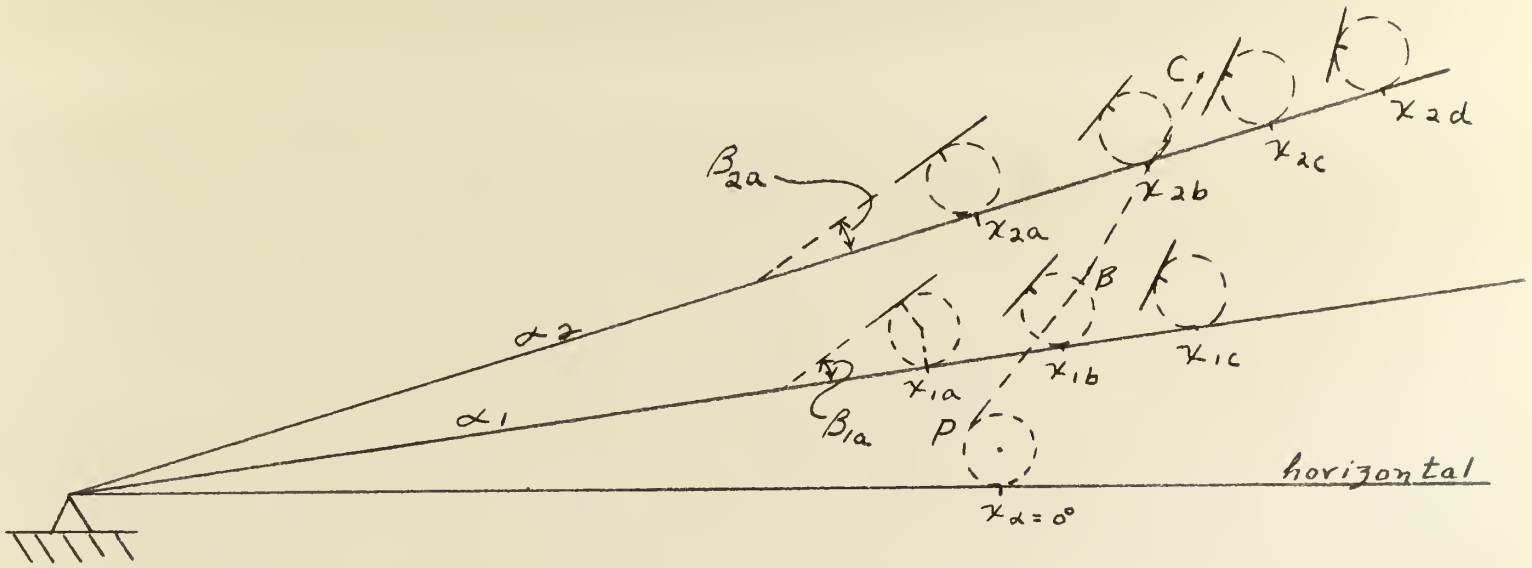


FIG. 16 SCHEMATIC OF GRAPHICAL SOLUTION

between the cam and cam bearing is on the lower half of the cam bearing for negative angles of alpha. In effect, there are two cams, one for positive values of alpha and another for negative values of alpha. There is a small difference in the values of beta near the indeterminate region at alpha equal to zero for these two cams. The resulting cam curves are shown full scale in Fig. 17 (inside back cover). Because of a bracing bar in the lever arm it was necessary to break away from the indicated curve at about minus 12 degrees for alpha, and from there on down the cam serves no purpose other than as a retainer guide for the weight suspension element.

Precision bearings, Boston Nice, radial ball bearings single row solid race type, No. 1616 DC were selected for both the cam bearings and for the lever arm contact bearings. This allowed a one-half-inch diameter shaft which was manufactured from drill rod.

Machining away material from the lever arm to provide the bearing track had greatly reduced the beam strength of the lever arm. It was necessary to reinforce this part of the lever arm in the manner indicated in Fig. 7. The beam strength was calculated for the new configuration to determine the maximum weight load which should be used. The results indicated that no more than 200 pounds-weight load should be used. This was considered adequate.

A support frame to hold the cam plates in position was designed and built as shown in Figures 10 and 11. The

use of slotted bolt holes allowed for some adjustment of the position of the cam plates within the support frame. The support members making the support frame rigid to the creep unit stand allowed for about two inches of fore and aft adjustment of the entire support frame and cam plates.

4. Performance of designed constant stress system.

The equipment was assembled and tested to determine the effective lever arm ratio for the starting position. The same starting position for successive tests was achieved by bringing the lever arm firm against the upper lever arm stop by means of a turnbuckle in the specimen mounting set up. A Baldwin SR-4 load cell, type U, with a Baldwin Lima Hamilton, type M, SR-4 strain indicator were used to give the value of F. The load cell and the strain indicator had a linear response of 4.0 micro inches per inch per pound. With all lost motion taken out of the specimen mounting set up, to the best judgement of the operator, the weight load was gradually applied by lowering a hydraulic jack which had been supporting the weights. Any lost motion which had not been taken out by the turnbuckle and any elastic deformation as the load was applied would allow some small movement of the lever arm from the starting position thereby decreasing the lever arm ratio. It was desirable to know what the effective lever arm ratio at starting position would be in order to know how much weight to use to achieve some desired stress. The results of the tests to determine the effective lever arm ratio at the starting position were as indicated in Fig. 18. Fairly wide scattering of the data points resulted from these tests. This scatter was believed due mostly to the inability of the operator to achieve the same initial tension in the turnbuckle on successive tests. This means that it may be

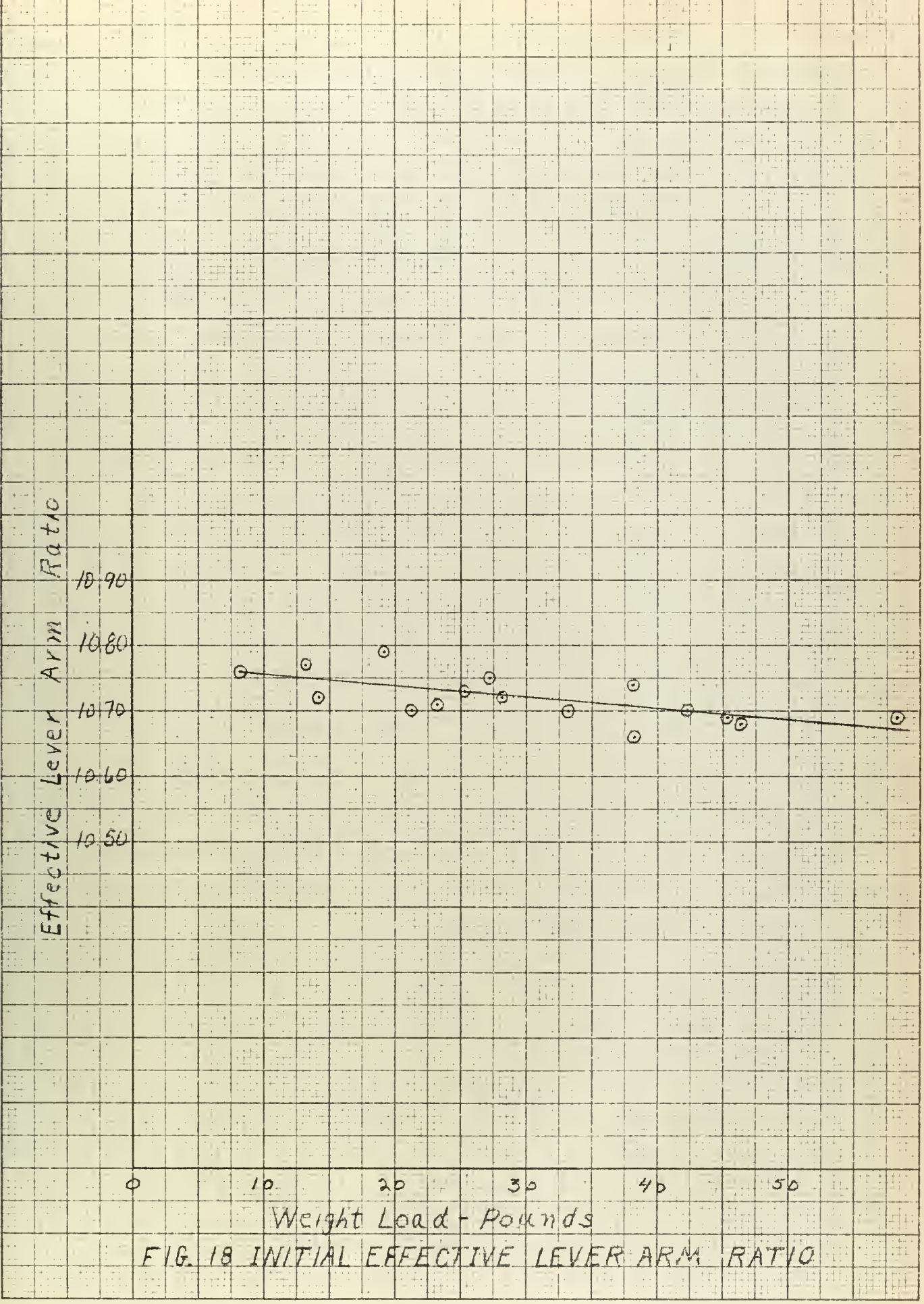


FIG. 18 INITIAL EFFECTIVE LEVER ARM RATIO

difficult to achieve a particular desired stress. However, whatever initial stress is achieved will be held constant.

Two tensile creep tests were run solely for the purpose of checking to what extent the system would keep the stress constant during the deformation of a specimen. Recrystallized low carbon steel was used at a temperature of 605°C. Very frequent readings of the dial extensometer and load cell strain indicator were taken. From these values the ratio F/F_0 could be determined. The ratio A/A_0 was picked off of Fig. 6. Then:

$$\frac{\sigma}{\sigma_0} = \frac{F/F_0}{A/A_0}$$

The values of σ/σ_0 were plotted versus the Δ gage values (Fig. 19). The results indicated that the stress was held to within one percent up to Δ gage of 0.75 inches and within two percent up to the maximum deformation of approximately 1.0 inches.

An unforseen obstruction of the weight suspension element at about minus nine degrees for alpha had led to the system being set up with the $\chi_{\alpha=0}$ slightly greater than was believed to be the designed value. After the first test the cam plates and support frame were moved a little closer to the creep unit stand just to observe the effect of the change in $\chi_{\alpha=0}$. The surprisingly good results at $x_{\alpha=0} = 29.8$ inches and the apparent worsening of results as $\chi_{\alpha=0}$ was shortened led to a careful review of the calculations which had gone into the cam design. It was discovered that a value of .725 for A/A_0 at alpha equal to zero had been used. This value

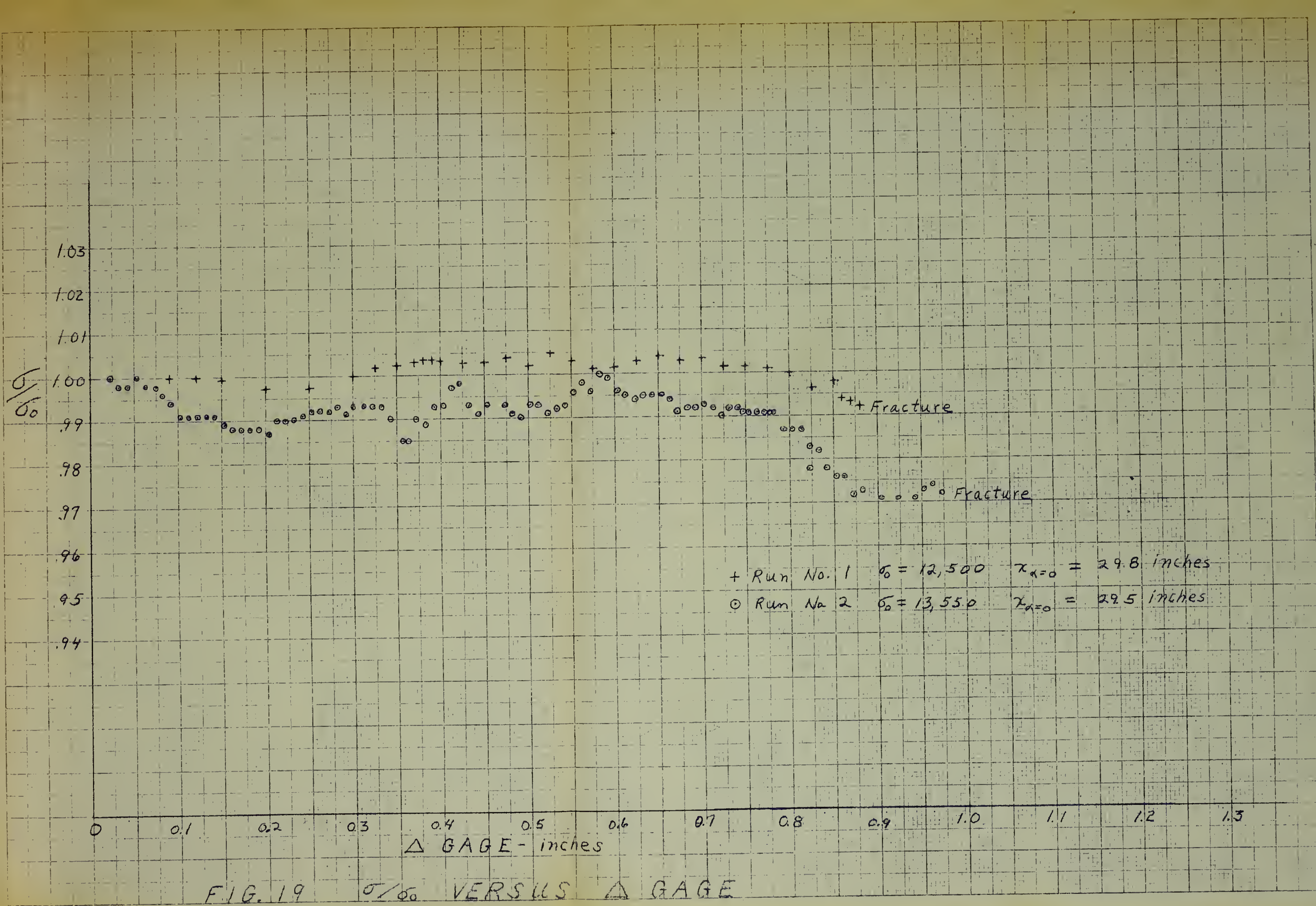


FIG. 19 σ/σo VERSUS Δ GAGE

goes into equation (8) and when divided into $\chi_{\alpha=0}$ produces the constant in the formula for x, equation (9). The value used for this constant had been 40.02. This means the actual $\chi_{\alpha=0}$ designed for was (40.02)(.741) or 29.6 inches. As this was the only place in the design calculations the value of $(A/A_0)_{\alpha=0}$ had been used the only effect of the error was to shift the designed value of x for alpha equal to zero.

The cam plates and support frame were adjusted back to the position such that $x_{\alpha=0} = 29.8$ inches for the subsequent metallurgical testing because this position gave best results.

5. The application of the constant stress system to creep testing.

The design and construction of a constant stress lever arm was prompted by some results recently obtained by Goldberg (6) at the U. S. Naval Postgraduate School in which low carbon steel was creep tested under constant load conditions. Hardness tests at room temperature were made on fractured creep specimens, measurements being taken along the gage section. These results are reproduced, schematically, in Fig. 20. The significant feature is that the fracture hardness was independent of the initial state (recrystallized, 15-percent cold worked, and50-percent cold worked) and creep test temperature and dependent only on the initial stress. This was somewhat surprising as fracture strains at the different test temperatures and for different initial specimen conditions for a given initial stress varied considerably, resulting in corresponding differences in fracture stress.

Hardness measurements on interrupted constant load tests showed that the hardness leveled off at about 15 to 25 percent elongation, depending on the temperature and initial state of the material. This suggested that either the initial stress or the stress reached in this elongation range was the stress primarily responsible for the final hardness.

It was decided to compare the reported data for an initial stress of 30,000 psi with that of a constant stress

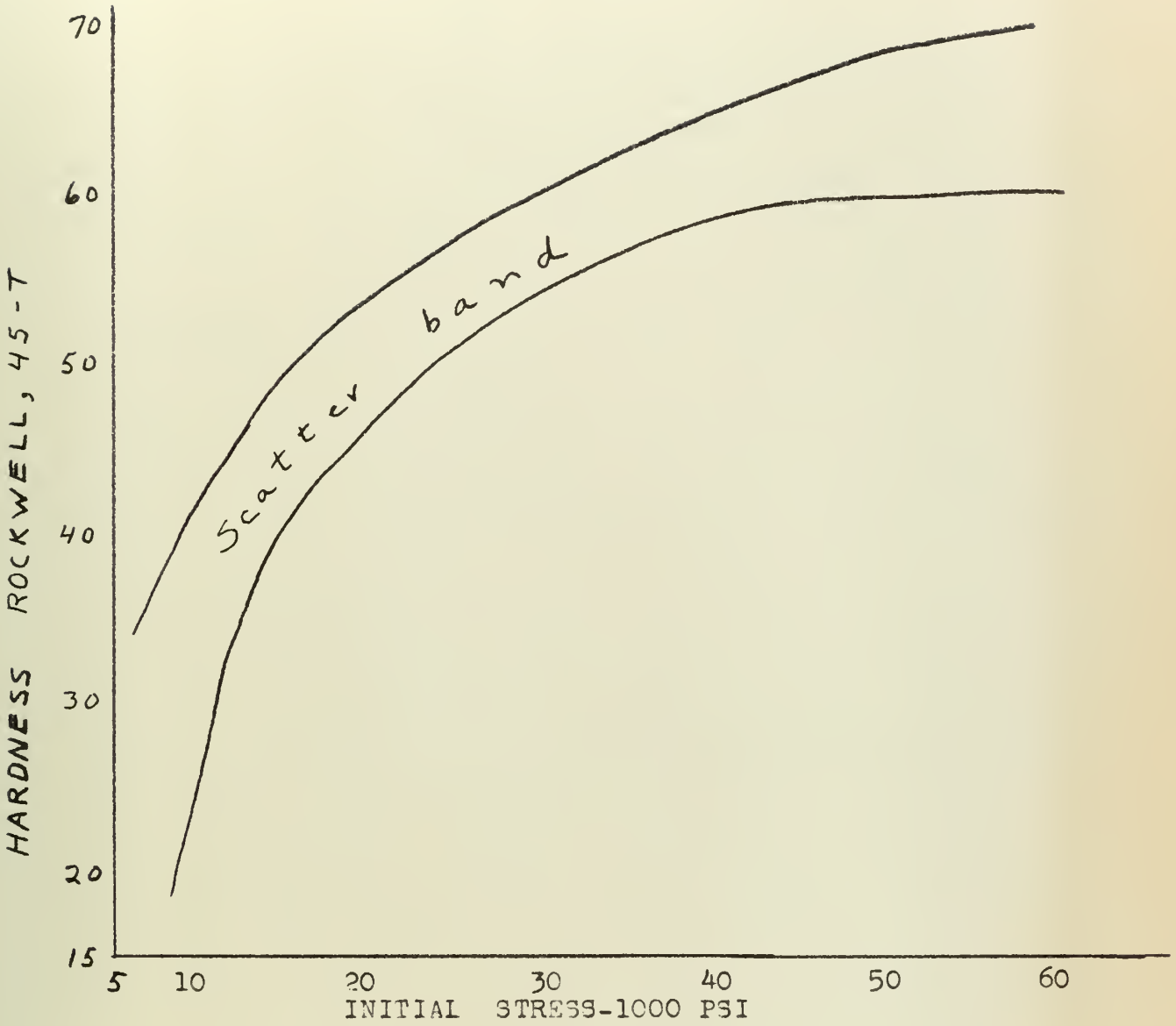


FIG.20 HARDNESS AT FRACTURE VERSUS INITIAL STRESS

of 30,000 psi and a constant stress of 36,000 psi, the latter corresponding to a nominal 20 percent elongation in the constant load test.

The same material was used for both studies, namely a 50-percent cold worked low carbon steel of the following composition: (Analysis by Kaiser Steel Company)

Element	C	Mn	P	S	Si	Cu	Ni	Cr
Percent	.06	.34	.010	.03	.01	.08	.02	.01

Element	V	Mo	Co	Al	Fe
Percent	.005	.005	.035	.012	Balance

Two other constant stress tests were run, one at 30,000 psi and one at 35,700 psi, and comparison between the basic creep curves for constant load and constant stress is presented in Fig. 21.

Interrupted constant stress creep tests were made for both 30,000 psi and 36,500 psi whereby the tests were frequently interrupted for room temperature hardness measurements. These results are presented in Fig. 22 together with the reported data for the constant load test at 30,000 psi of the same material.

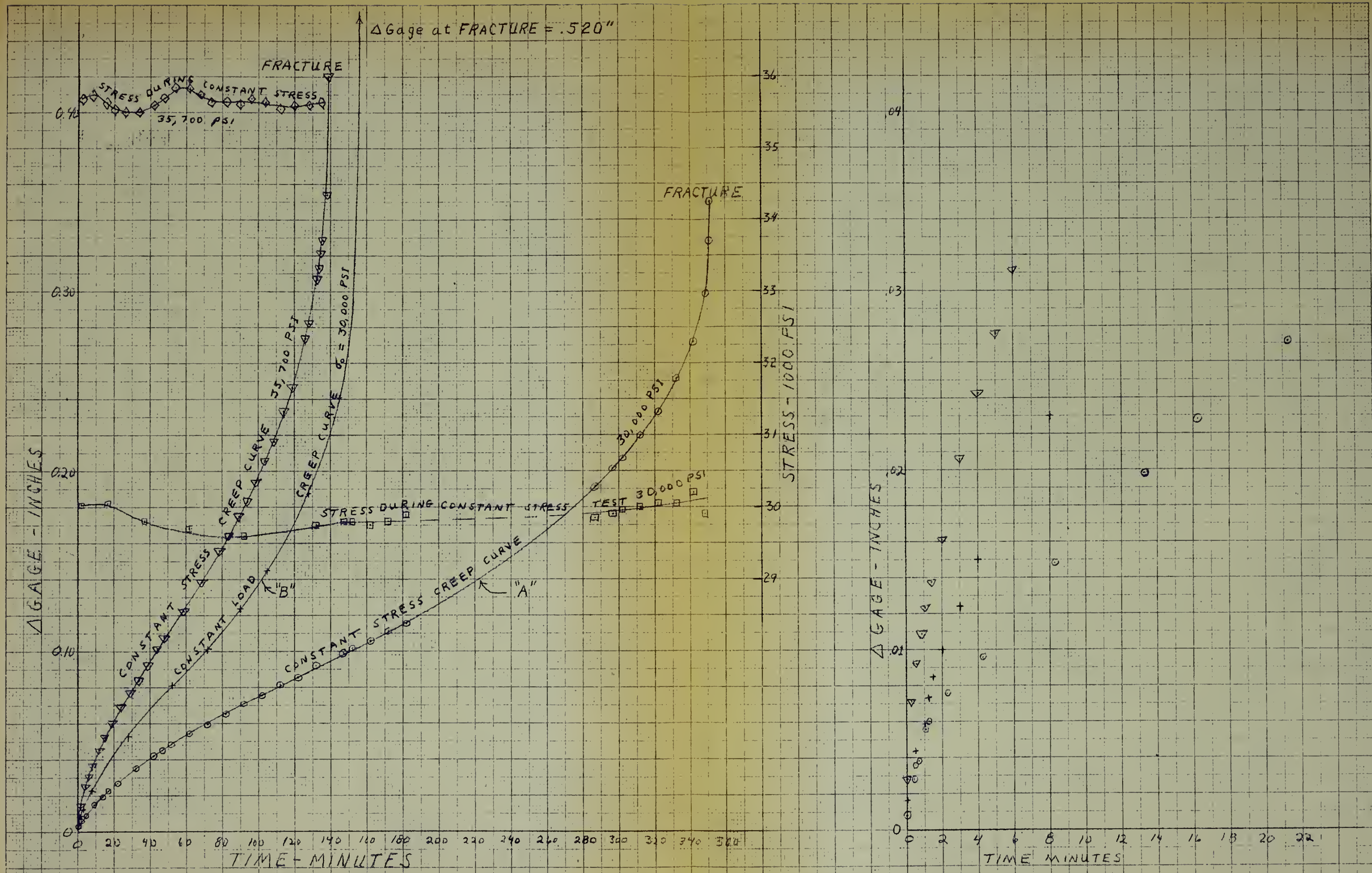


FIG. 21 CONSTANT STRESS AND CONSTANT LOAD CREEP CURVES

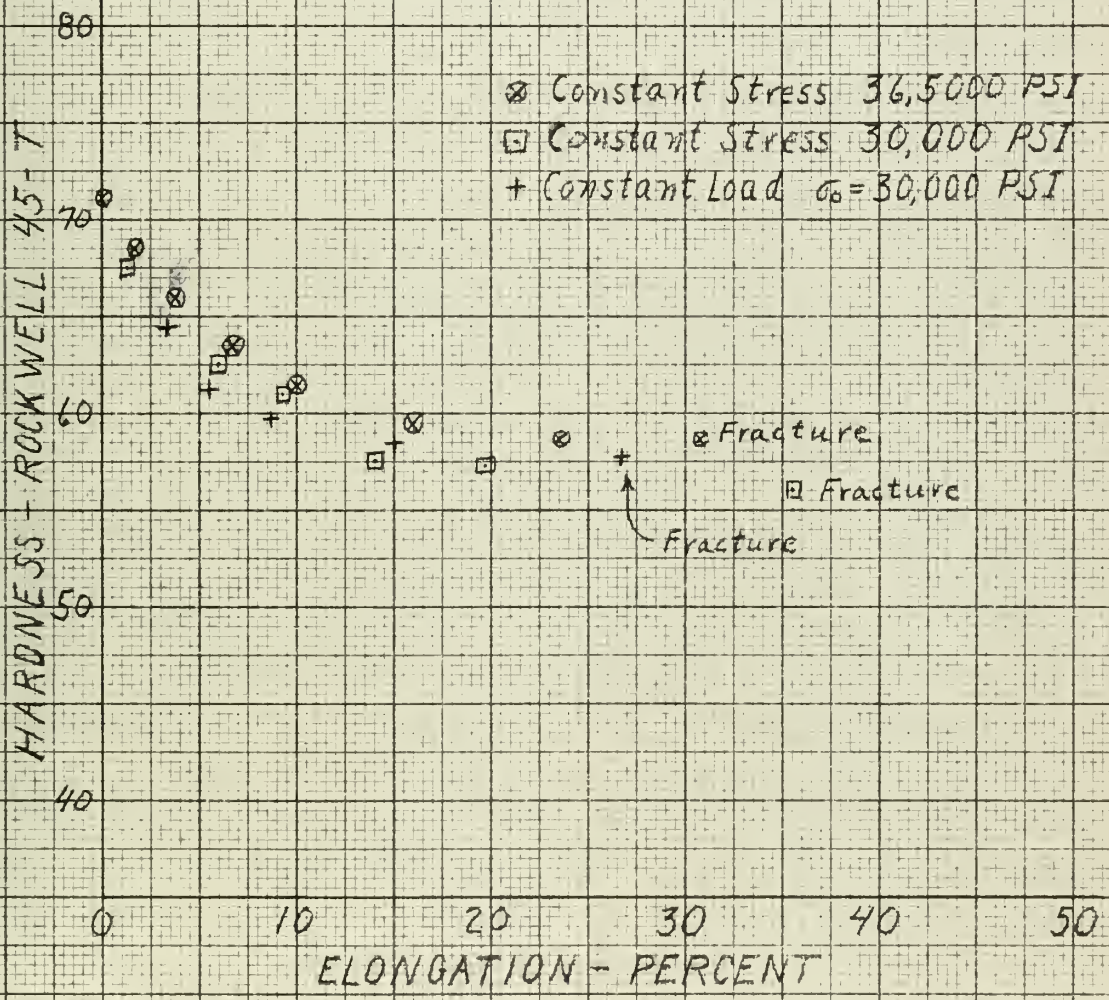


FIG. 22 HARDNESS VERSUS ELONGATION

6. Interpretation of creep results.

The relative displacement ofthe basic creep curves, shwon in Fig. 21, is as may be expected; namely, the larger instantaneous creep stress leads to more rapid creep rates. Thus, for corresponding strains, the creep rate is lowest for the 30,000 psi constant stress curve, throughout the entire strain range, except for the last instant before fracture, which is generally associated with severe necking. A comparison between the 30,000 psi constant load and the 35,700 psi constant stress curves shows a reversal in relative creep rates at about 10 percent elongation. By plotting the difference in time required to reach a given value of $\triangle$ gage against $\triangle$ gage, as shown in Fig. 23, this reversal in relative creep rates between the 30,000 psi constant load and the 35,700 psi constant stress curves was found to be at a $\triangle$ gage of .16 inches. From Fig. 6 it was calculated that a value of $\triangle$ gage of .322 corresponds to the point at which the instantaneous stress of the constant load test became equal to the 35,700 psi stress of the constant stress test. This means that at a smaller instantaneous stress the constant load specimen had a strain rate equal to or greater than that of the specimen at a constant stress of 35,700 psi, which indicates that the constant load specimen had a smaller resistance to creep at that strain.

The interrupted creep test hardness data for the different stress conditions, as shown in Fig. 22, appear to

FIG. 23 PLOT OF TYME DIFFERENCE FOR SAME STRAIN BETWEEN 30,000 PSI CONSTANT LOAD AND 35,700 CONSTANT STRESS VERSUS Δ GAGE

fall within a normal scatter band which levels off at about 15 to 20 percent extension. However, the values for the 36,500 psi test are consistently higher and level off at a higher value. Thus, the strain softening for a cold-worked material would appear to be dependent on the initial stress and the amount of deformation. This is consistent with the basic creep data for it indicates that more strain softening at the same deformation occurred in the constant load specimen where the initial stress was 30,000 psi than occurred in the 35,700 psi constant stress specimen. This greater softening allowed the strain rate to become greater for the constant load specimen even though the stress at that point was still smaller than the constant stress specimen.

Admittedly, the conclusions drawn are based on somewhat meager data. Experiments involving a change of stress from a constant stress curve to a value corresponding to a stress for the same deformation on a constant load curve with the same initial stress (or visa versa) should prove to be critical tests. For example, with reference to Fig. 21, if the stress were suddenly changed from point "A" to point "B", then the new strain rate should correspond to the instantaneous value at "B" on the constant load curve.

7. Conclusions and recommendations.

A creep unit adaption was designed and built which gave a stress constant to within plus or minus one percent over a range of 66 percent extension.

The system represents a very economical and space saving method of producing stress of acceptably small variations. The early and wide variation of the minimum cross-sectional area from that predicted by assumption of constant volume makes greater accuracy virtually impossible for any system which does not include some sort of continuous measuring of cross-sectional area during testing.

Some difficulty in exactly achieving a particular stress was experienced due to the slight variation with load of the effective lever arm ratio in the starting position. It is recommended that some type of load cell always be used in order that the exact stress produced may be known.

It appears that the strain softening for a given material and temperature depends only on the initial stress and amount of deformation. Further critical tests have been recommended to verify this conclusion.

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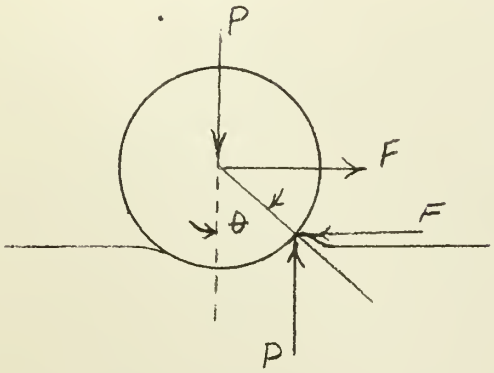
APPENDIX I

An estimate of the effect of bearing friction.

For rolling action the situation is, in exaggerated form:



$F\ r \cos\theta = P\ r \sin\theta$ where r = radius.

and

$$F\ r = P\ r \tan\theta$$

The term $r \tan\theta$ is conventionally called the coefficient of friction f_r .

Then:

$$f_r = r \tan\theta$$

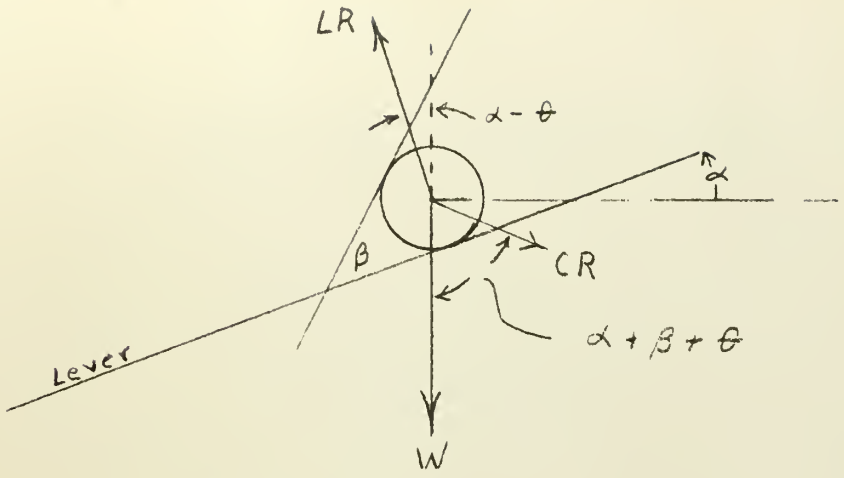
$$\theta = \tan^{-1} \frac{f_r}{r}$$

Taking the most conservative value of f as .0015 and the radius equal to 0.5 inches

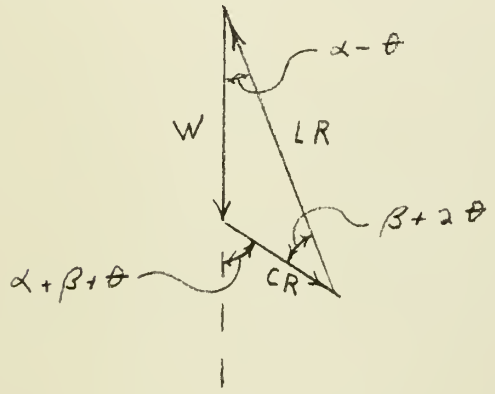
$$\theta = \tan^{-1} \frac{.0015}{0.5}$$

$$\theta = .003 \text{ radians}$$

For the situation:



where: CR = cam reaction
LR = lever reaction



$$LR = W \frac{\sin(\alpha + \beta + \theta)}{\sin(\beta + 2\theta)}$$

If there were no rolling friction:

$$\overline{LR} = W \frac{\sin(\alpha + \beta)}{\sin \beta}$$

Therefore

$$\frac{LR}{\overline{LR}} = \frac{\sin\beta \sin(\alpha + \beta + \theta)}{\sin(\alpha+\beta) \sin(\beta + 2\theta)}$$

$$\frac{LR}{\overline{LR}} = \frac{\sin\beta\left[\sin(\alpha+\beta)\cos\theta + \cos(\alpha+\beta)\sin\theta\right]}{\sin(\alpha+\beta)\left[\sin\beta\cos 2\theta + \cos\beta\sin 2\theta\right]}$$

$$= \frac{\sin\beta\cos\theta + \sin\beta\sin\theta\cot(\alpha+\beta)}{\sin\beta\cos 2\theta + \cos\beta\sin 2\theta}$$

$$= \frac{\cos\theta + \dfrac{\sin\theta}{\tan(\alpha+\beta)}}{\cos 2\theta + \dfrac{\sin 2\theta}{\tan\beta}}$$

$$LR/\overline{LR} \approx \frac{1 + \dfrac{\theta}{\tan(\alpha+\beta)}}{1 + \dfrac{2\theta}{\tan\beta}}$$

$$\approx 1 + \frac{\theta}{\tan(\alpha+\beta)} - \frac{2\theta}{\tan\beta}$$

$$\approx 1 + \theta\left(\frac{1}{\tan(\alpha+\beta)} - \frac{2}{\tan\beta}\right)$$

$LR/\overline{LR}$ will very with α and β. It was anticipated that β would be close to 45°.

Then: $$LR/\overline{LR} \approx 1 + \theta\left(\frac{1-\tan\alpha}{1+\tan\alpha} - 2\right)$$

for $\alpha = 10°$ $$LR/\overline{LR} \approx 1 + .003\left(\frac{.8237}{1.1763} - 2\right)$$

$$\approx 1 - .003(1.30)$$

$$LR/\overline{LR} \approx 1 - .0039$$

Note that LR is not perpendicular to the lever arm if rolling friction is considered. The moment of LR about the pivot knife edge would be:

$$moment = LR \times \cos \theta$$

However, as shown , $\cos\theta = 1$, therefore, no further error is introduced by considering

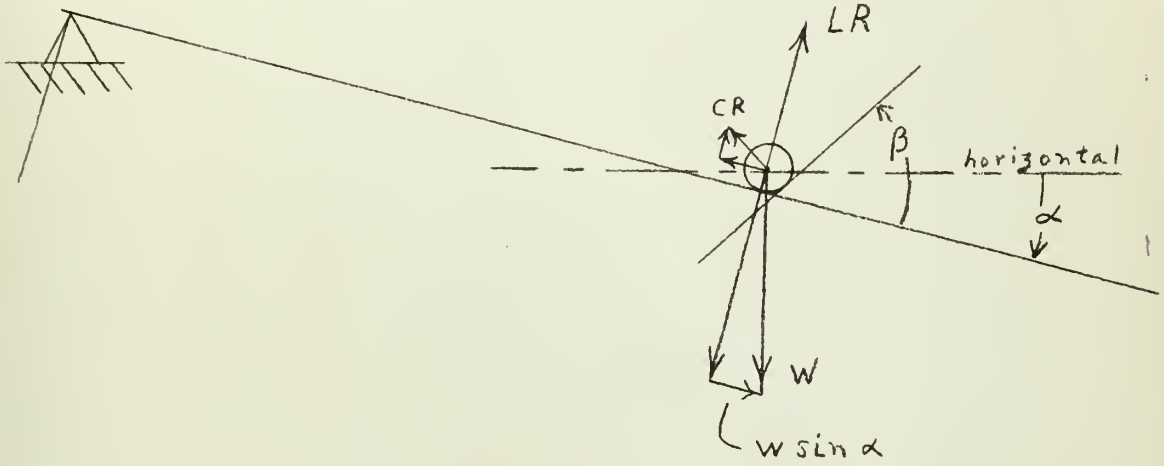
$$moment = LR \times$$

and the maximum error due to rolling friction will be about 0.38 percent.

APPENDIX II

Solution for lever below the horizontal

With the lever arm in a position below the horizontal the forces on the bearings and bearing shaft will be:



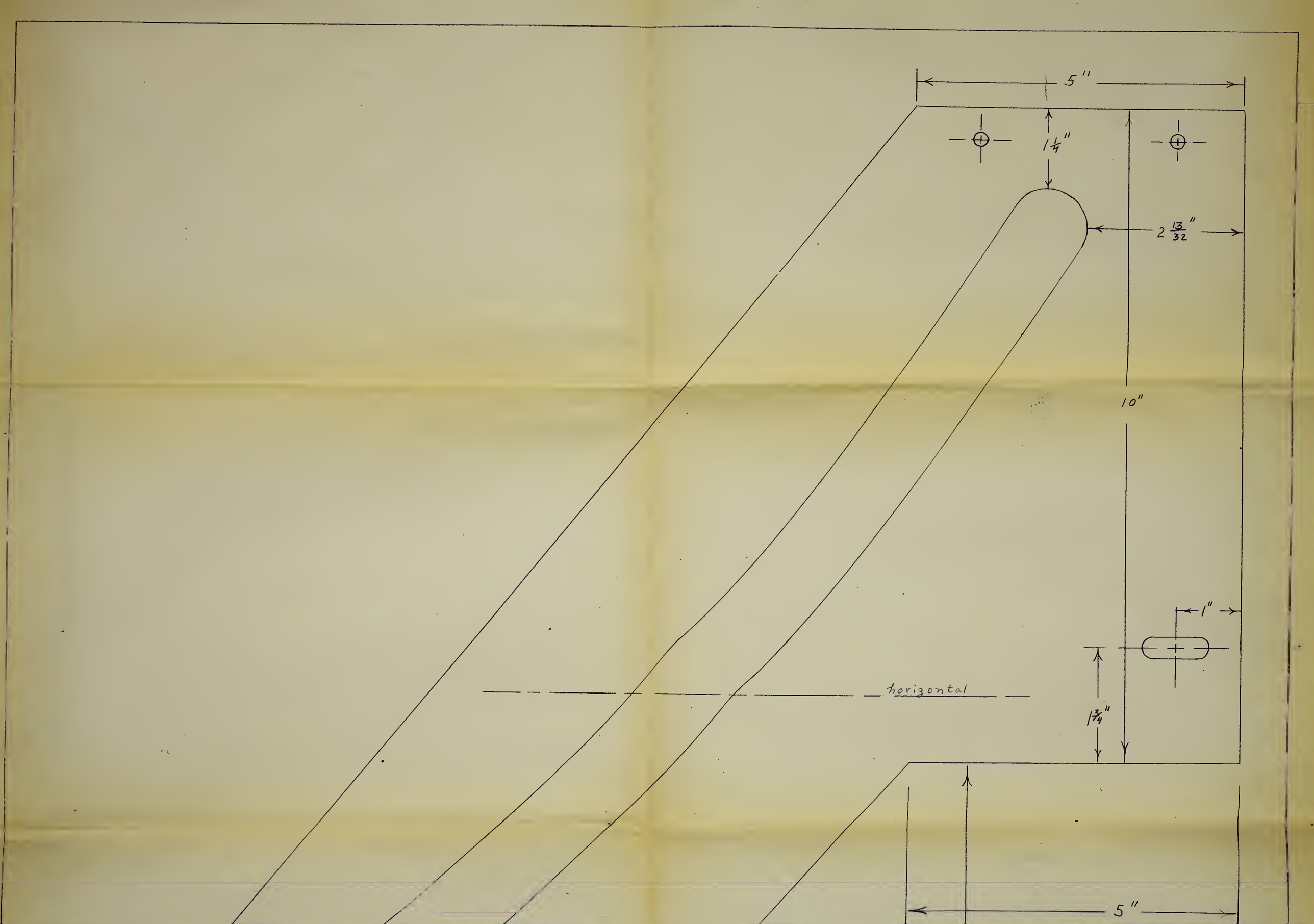
W resolves into W sinα along the track and W cosα perpendicular to the track. As before, the cam reaction along the bearing track must be W sin α . Therefore, the component of cam reaction perpendicular to the bearing track is

$$W \frac{\sin\alpha}{\tan\beta}$$

Therefore:

$$\text{Lever Reaction, } LR = W\left(\cos\alpha - \frac{\sin\alpha}{\tan\beta}\right)$$

Which is the same as for positions of the lever above the horizontal except for he negative sign of the cam term. Therefore, equation (1) is valid for lever positions below the horizontal if , measured as indicated above , is read as a negative angle.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1¼"
2 13/32"
10"
1"
1¾"
horizontal
5"

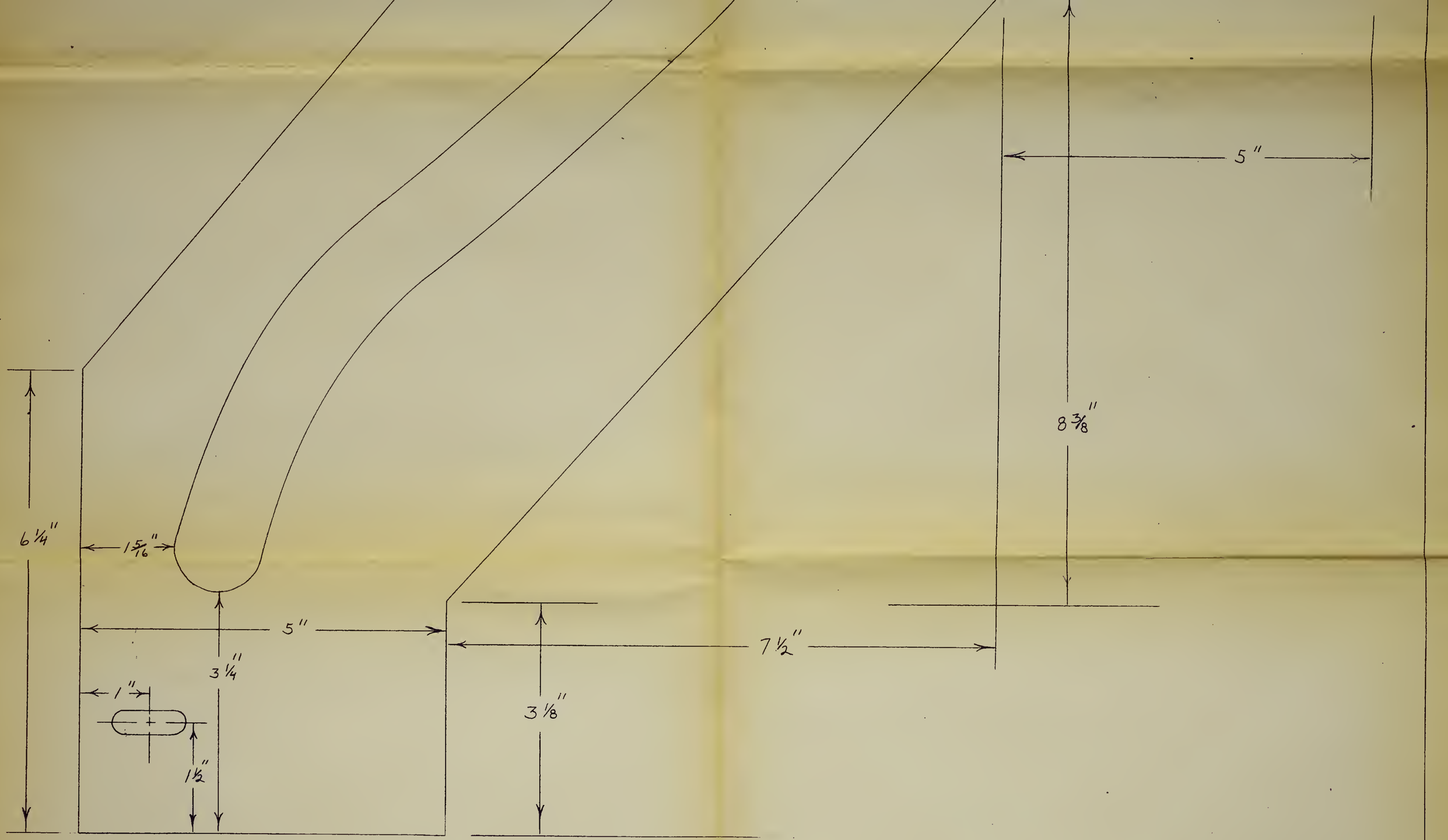

FIG. 17 CAM CURVE AND CAM PLATE, FULL SCALE DRAWING

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